VISIT…

ROLLING ELEMENT BEARINGS

W. J. Derner and E. E. Pfaffenberger

ROLLING BEARING TYPES

Rolling contact bearings are generally categorized by the type of rolling element and the manner in which it is used. The most obvious divisions are between ball and roller bearings and between radial and thrust bearings, while some angular contact bearings are utilized for both radial and thrust loads.

Individual sections in this chapter will cover (1) types of rolling element bearings and their selection criteria, (2) dimensional standards, (3) characteristics of materials employed, (4) rolling bearing theory, (5) load, speed, and related application limits, (6) lubrication, and (7) failure analysis. Load rating for individual bearings, their application ranges, and many other details are available in catalogs and related literature from bearing suppliers.

Ball Bearings (Figure 1)

Radial and Angular Contact

The most common design of radial ball bearing is the Conrad type where, in general, five to nine balls are inserted between the inner and outer rings. Where a greater radial capacity is desired, filler type rings utilize notches on one shoulder so that more balls can be inserted. An optimum capacity is achieved through the use of a split inner or outer ring wherein a maximum number of balls can be inserted with the retainers. This latter design, however, requires external means for holding the ring halves together so that load can be divided between contacts on both halves.

Self-aligning ball bearings are available in double row varieties in which the outer ring raceway is of a larger radius than the ball and the inner has two raceways ground in it, one for each row of balls.

Thrust Ball Bearings

These are available in designs for single direction as well as double direction thrust and normally are found with 90° contact angles.

Roller Bearings

A variety of rollers have developed for use in bearings including the early spring-wound cylindrical, solid and hollow cylindrical, tapered, and spherical rollers. Retainers (cages) used to space the rolling elements may be land riding, roller riding, or supported on a raceway. Retainers may be (1) machined out of solid, (2) stamped and formed, (3) fabricated and fastened by cold heading or riveting, or (4) molded of one or two elements. Since retainer contact with the roller or ring involves some sliding, a lubricant should be chosen which is compatible with the nature of the contact as well as the material of the retainer.

Radial and Angular Contact (Figure 2)

Tapered roller bearings — Supported between two cones of different angles, the tapered roller centers itself between them and recognizes a certain axial force which maintains its contact with a lip or rib generally on the inner ring. Its contact with that rib involves sliding which must be lubricated to prevent wear and to dissipate the heat generated.

Cylindrical roller bearings — This type generally runs cooler than other roller bearings because of the narrow and uniform shape of the Hertzian contact with no more roller end contact with the ribs or flanges than is required to provide guidance or location. In high-

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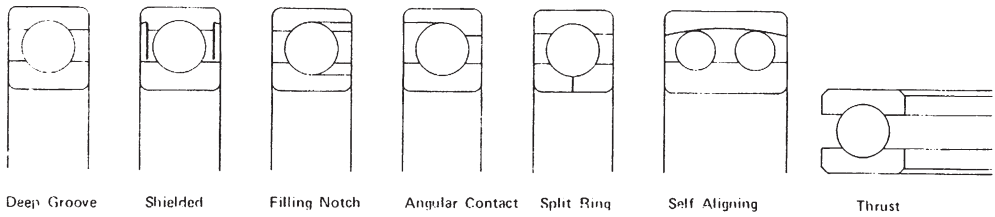

FIGURE 1. Common types of ball bearings.

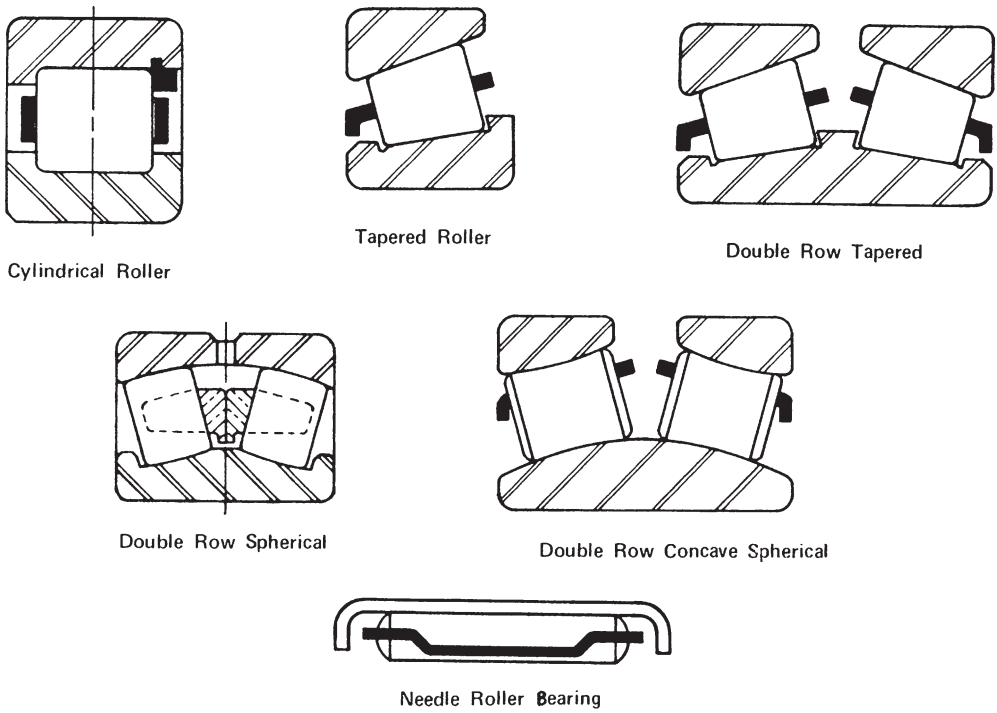

FIGURE 2. Common types of radial roller bearings.

speed application, positive guidance is essential to reduce parasitic losses due to the working of the roller back and forth between the ribs.

Spherical roller bearings — These are produced in both asymmetrical and symmetrical roller designs which have their unique requirements for lubrication and cooling. The design with asymmetrical rollers has a flange contact similar to that in tapered roller bearings and requires careful lubrication. The symmetrical roller design is supplied either with a retainer guided roller or with stabilizing rings which utilize the retainer as a spacer or separator only. These latter two spherical bearings are comparable in their performance and generally are utilized in applications which require greater load carrying capacity and/or run at higher speeds. Concave roller designs are generally produced with stamped steel retainers and are sensitive to loading variations which may produce undesirable skewing of the rollers. Because of the reliance on formed steel retainers for guidance, these bearings generally run warmer than those with convex rollers.

Thrust Bearings (Figure 3)

Tapered roller bearings — Tapered roller bearings with contact angles of 45° and more are used for heavy duty thrust applications that include slow speed, high shock load appli-

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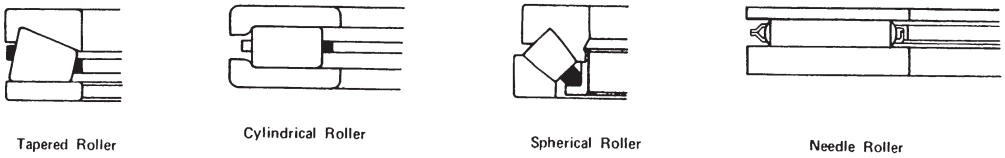

FIGURE 3. Roller thrust bearings.

cations similar to those in screw-down bearings in rolling mills. Since many of these run at low speeds or are even subjected to only momentary, limited rotation (and/or oscillation), lubrication requirements are highly specialized and require careful evaluation.

Cylindrical roller bearings — Conventional roller thrust bearings with flat plate surfaces are utilized in low-speed applications where high thrust capacity is essential. Tandem cylindrical roller thrust bearings are used in many applications wherein single-helical gears are run at low speeds. Because of the relatively high slip in these applications, as well as the generally severe contact conditions in all slow speed and/or oscillating operation, special attention to cooling and extreme pressure (EP) additives in the lubricant are required.

Spherical roller bearings —- Spherical thrust bearings require the same attention as do most tapers, but absorb misalignments which would not be acceptable to cylindrical and tapered varieties. Spherical thrust bearings are provided with both symmetrical and asymmetrical roller designs. The latter are somewhat limited in having a moderately loaded rib or locating flange as a sliding contact which must be lubricated as with the tapered roller bearings. In lower-speed applications where hydrodynamic films are not readily generated at the rib roller end contact, wear becomes a significant factor limiting the life or function of these bearings. In all cases of angular contact, spherical or tapered roller thrust bearings, the relation of thrust to radial load must be carefully controlled to ensure that the bearing does not come apart. Manufacturers' recommendations must be carefully adhered to.

Mounted-Bearing Units

A significant proportion of rolling element bearings are supplied in integral housings with seals which offer advantages in that they do not require a large, continuous machined housing. While a great majority are grease lubricated, for adverse environmental conditions some are provided with complete lubricating systems to include cooling and filtration. For severe contamination, flushable seals (Figure 4) have lube fittings separated from the main lubrication system of the bearing. Some applications are so severe that frequent and heavy relubrication is relied on to purge the system of contaminants and to exclude water vapors due to "breathing" where intermittent operation is encountered.

ROLLING BEARING STANDARDS

Boundary Dimensions

The great majority of the world rolling element bearings are in compliance with boundary dimension plans adopted by the International Standards Organization (ISO). Domestic standards originated with the Anti-Friction Bearing Manufacturer's Association (AFBMA) have been taken over by American National Standards Institute (ANSI) which, as a participating body in ISO work, offers the most complete and authoritative set of standards for use in this country.

A great majority of the so-called inch series of bearings have been superceded by metric series which fit a number of well-established boundary plans.[1] Of particular note, a new series of tapered roller bearings has achieved a reduction in the multiplicity and complication of sizes with worldwide acceptance in an ISO standard.[2]

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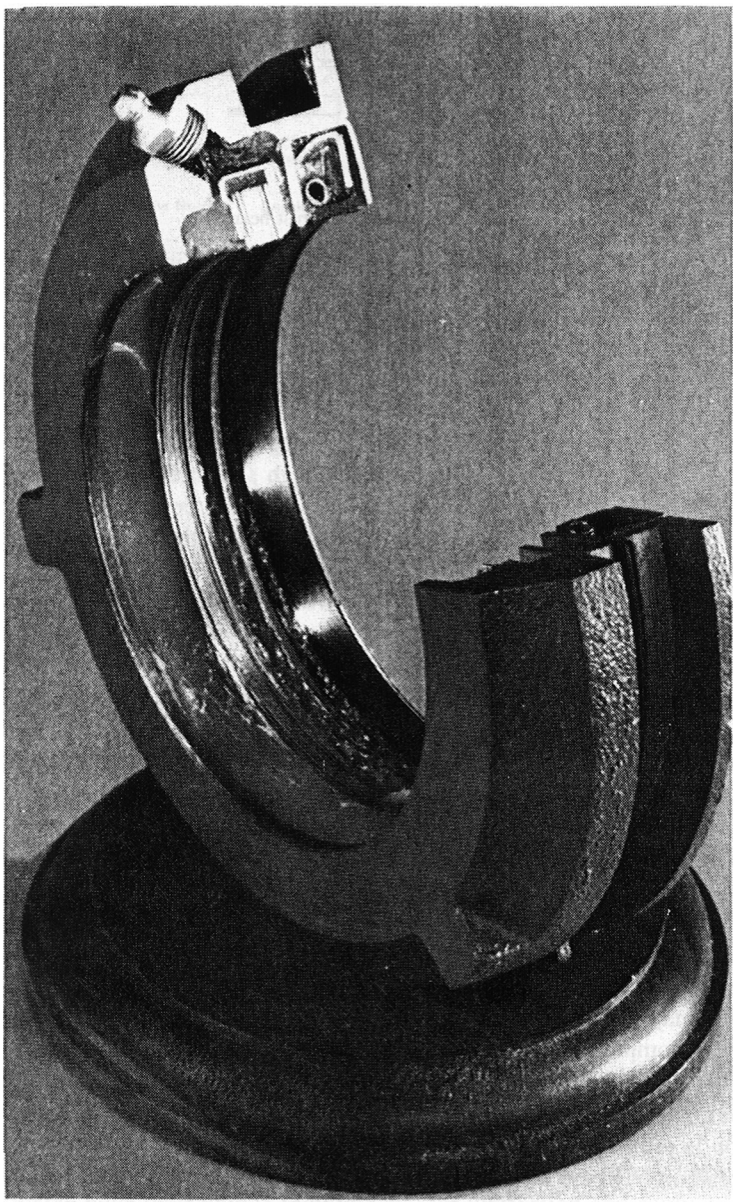

FIGURE 4. Flushable seals with lubricant fittings for severe contamination.

As will be seen in Figure 5, the basic metric series cover bore, OD, and width and fully describe the annuli for each increment of bore size. There are some exceptions, but over the range of 15 to 130 mm most series have bores with 5-mm increments. Above 130 mm increments are normally 10 mm or larger.

Most bearings sold in this country offer dimensions which are inch equivalents of even millimeters. For a full description of all standard bearings which are currently manufactured, it is recommended that ANSI be contacted for the current standards and then refer back to manufacturer's catalogs for availability of bearings fitting these boundary dimension series.

While it is possible to justify the design, development, manufacturing, and application of bearings which do not meet the standard boundary dimension plans, application of special

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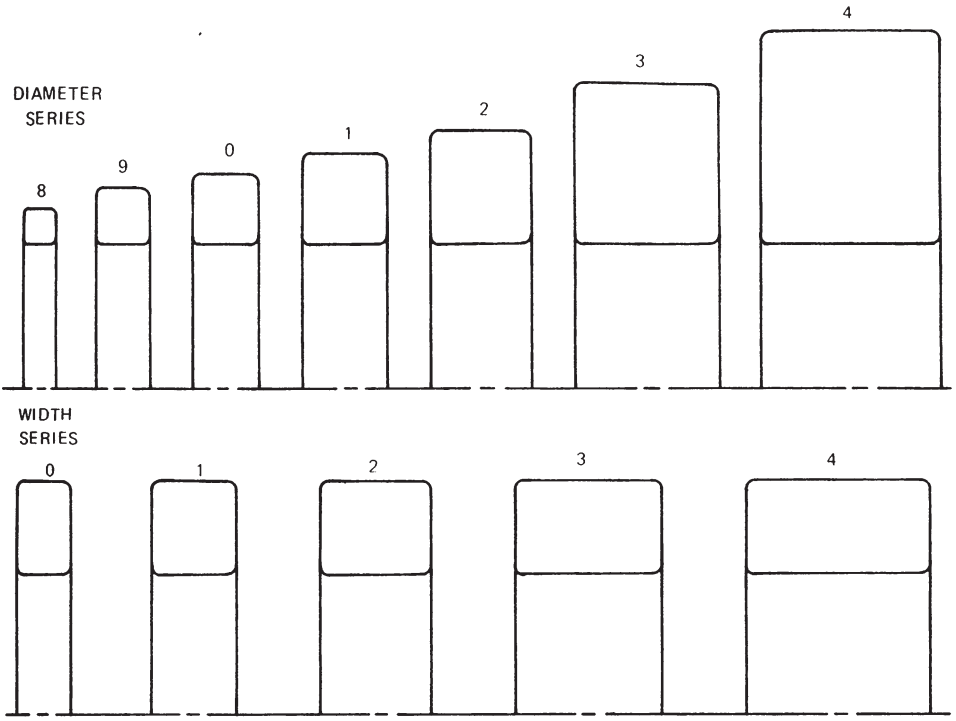

FIGURE 5. Graphical representation of the diameter and width series for radial bearings. (From *Metric Ball and Roller Bearings Conforming to Basic Boundary Plans,* ANSI/AFBMA Standard 20, Anti-Friction Bearing Manufacturers Association, Arlington, Va., 1977. With permission.)

designs requires careful consideration and prior evaluation. The useful life of most machinery is often extended by overhaul, at which time replacement of bearings may be essential. At overhaul the availability of new bearings may be a real problem and any selection of a bearing which does not fit standard published boundary dimensions must include a plan for resupply. In some industries the replacement bearings are procured at the time of original manufacture and then are stocked either by the user or the supplier. Machines manufactured in higher volumes quite often will employ special bearings designed or selected by the manufacturer and stocked by his dealers around the country. Typical would be the automotive service parts groups.

Because the wide variety of boundary plans, it is suggested that special bearings be considered only when weight or size limitations are a problem.

Internal Clearances

Also available through ANSI are standards for the clearance ranges for all sizes of bearings as shown in Tables 1 to 3. While the standard clearance range is normally designated as symbol 0. some manufacturers have their own nomenclature. Reference to bearing manufacturers' catalogs and sales literature will relate exact clearance ranges to requirements of applications for each type of bearing. It is desirable to coordinate quality control procedures with the supplier to ensure a minimum of confusion and delay in qualifying bearings for internal clearance.

BEARING MATERIALS

The four essential properties of bearing material—strength, fatigue resistance, wear resistance, and dimensional stability will be considered separately.

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Table 1
RADIAL INTERNAL CLEARANCE, SINGLE-ROW, RADIAL CONTACT, BALL BEARINGS

Tolerance Limits in 0.0001 In.

Basic bore diameter (mm)		Symbol 2[a]				Symbol O (normal)[a]				Symbol 3[a]				Symbol 4[a]			
		Acceptance limits															
		Manufacturing limits				Manufacturing limits				Manufacturing limits				Manufacturing limits			
Over	Incl.	Low	Low	High	High	Low	Low	High	High	Low	Low	High	High	Low	Low	High	High
2.5	10	—	—	2.5	3.0	1	1.5	4.5	5	3	4	8	9				
10	18	—	—	3.0	3.5	1	2	6	7	4	5	9	10	7	8	12	13
18	24	—	—	3.5	4.0	2	3	7	8	5	6	10	11	8	9	13	24
14	30	—	—	4	4.5	2	3	7	8	5	6	10	11	9	10	15	16
30	40	—	—	4	4.5	2	3	7	8	6	7	12	13	11	12	17	18
40	50	—	—	4	4.5	2	3	8	9	7	8	13	14	12	13	19	20
50	65	—	1	5	6	3	4	10	11	9	10	16	17	15	16	23	24
65	80	—	1	5	6	4	5	11	12	10	11	19	20	18	19	27	28
80	100	—	1	6	7	5	6	13	14	12	13	22	23	21	22	32	33
100	120	—	1	7	8	6	7	15	16	14	15	25	26	24	25	37	38
120	140	—	1	8	9	7	8	18	19	16	18	30	32	28	30	43	45
140	160	—	1	8	9	7	8	20	21	18	20	34	36	32	34	49	51
160	180	—	1	9	10	8	9	23	24	21	23	38	40	36	38	55	58
180	200	—	1	11	12	10	11	27	28	25	27	44	46	42	44	62	64

Note: Tolerance limits for radial internal clearance of single-row, radial contact ball bearings under no load (applicable to bearings of ABEC 1, ABEC 5, ABEC 7, and ABEC 9 Tolerance Classes).

[a] These symbols relate to AFBMA Standard Section 5 — Identification Code.

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Table 2
RADIAL INTERNAL CLEARANCE, CYLINDRICAL ROLLER BEARINGS

Tolerance Limits in 0.0001 in.

Basic bore diameter (mm)		Symbol 2[a]				Symbol O[a] (normal)				Symbol 3[a]				Symbol 4[a]			
		Interchangeable[b]															
			Matched[c]				Matched[c]				Matched[c]				Matched[c]		
Over	Incl.	Low	Low	High	High	Low	Low	High	High	Low	Low	High	High	Low	Low	High	High
	10	0	4	8	12	4	8	12	16	10	14	18	22	14	18	22	26
10	18	0	4	8	12	4	8	12	16	10	14	18	22	14	18	22	26
18	24	0	4	8	12	4	8	12	16	10	14	18	22	14	18	22	26
24	30	0	4	10	12	4	10	14	18	12	16	20	26	16	20	24	28
30	40	0	5	10	14	6	10	16	20	14	18	22	28	18	22	28	32
40	50	2	6	12	16	8	12	18	22	16	20	26	30	22	26	32	35
50	65	2	6	14	18	8	14	20	26	18	22	30	35	26	30	35	41
65	80	2	8	16	22	10	16	24	30	22	28	35	41	30	35	43	49
80	100	4	10	18	24	12	18	28	32	26	32	41	45	35	41	49	55
100	120	4	10	20	26	14	20	32	35	32	37	47	53	41	47	57	63
120	140	4	12	24	30	16	24	35	41	35	41	53	61	45	53	63	71
140	160	6	14	26	32	20	26	39	45	39	45	59	65	51	59	71	77
160	180	8	14	30	34	24	30	43	49	43	49	65	69	59	65	79	85
180	200	10	16	32	37	26	32	47	53	49	55	71	77	65	71	87	92
200	225	12	18	35	41	30	—	—	59	55	—	—	85	71	—	—	100
225	250	16	20	39	45	35	—	—	65	61	—	—	90	81	—	—	110
250	280	18	22	43	49	39	—	—	71	69	—	—	100	90	—	—	122
280	315	20	24	47	52	43	—	—	77	77	—	—	110	100	—	—	134
315	355	22	26	53	57	49	—	—	85	85	—	—	120	110	—	—	146
355	400	26	30	59	63	55	—	—	93	96	—	—	134	126	—	—	163

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Table 2 (continued)
RADIAL INTERNAL CLEARANCE, CYLINDRICAL ROLLER BEARINGS

Tolerance Limits in 0.0001 in.

Basic bore diameter (mm)		Symbol 2[a]				Symbol O[a] (normal)				Symbol 3[a]				Symbol 4[a]			
		Interchangeable[b]															
			Matched[c]				Matched[c]				Matched[c]				Matched[c]		
Over	Incl.	Low	Low	High	High	Low	Low	High	High	Low	Low	High	High	Low	Low	High	High
400	450	28	—	—	75	61	—	—	108	106	—	—	153	140	—	—	179
450	500	33	—	—	81	71	—	—	118	118	—	—	165	155			202

Note: Tolerance limits for radial internal clearance of cylindrical roller bearings under no load. This table applies to bearings of bore, O.D. and width dimensions conforming to AFBMA Standard Section 2, Table 1 and chamfer dimensions conforming to AFBMA Standard Section 3, Table 5.1 and 5.2

[a] These symbols relate to the AFBMA Standard Section 5 — identification code (edition later than December 1962).
[b] The term "interchangeable" refers to such assembly of rings and rollers that the separable ring *can* be replaced by any other ring of the same design and manufacture.
[c] The term "matched" refers to such assembly of rings and rollers that the separable ring *cannot* be replaced by any other ring.

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Table 3
RADIAL INTERNAL CLEARANCE SELF-ALIGNING ROLLER BEARINGS, METRIC

Tolerance Limits in 0.0001 In.

Basic bore diameter (mm)		Bearings with cylindrical bore							
		Symbol 2[a]		Symbol O[a] (normal)		Symbol 3[a]		Symbol 4[a]	
Over	Incl.	Low	High	Low	High	Low	High	Low	High
14	24	4	8	8	14	14	18	18	24
24	30	6	10	10	16	16	22	22	28
30	40	6	12	12	18	18	24	24	32
40	50	8	14	14	22	22	30	30	39
50	65	10	17	17	26	26	36	36	47
65	80	12	20	20	32	32	44	44	57
80	100	14	25	25	39	39	53	53	71
100	120	17	31	31	48	48	64	64	83
120	140	20	38	38	57	57	75	75	95
140	160	24	43	43	65	65	87	87	110
160	180	26	47	47	71	71	95	95	122
180	200	28	51	51	79	79	103	103	133
200	225	32	55	55	87	87	114	114	149
225	250	36	59	59	95	95	126	126	165
250	280	39	67	67	103	103	138	138	180
280	315	44	75	75	110	110	145	145	197
315	355	47	79	79	122	122	161	161	216
355	400	51	87	87	134	134	177	177	236
400	450	55	95	95	145	145	197	197	260
450	500	55	103	103	162	162	216	216	284
500	560	59	110	110	173	173	236	236	307
560	630	67	122	122	189	189	256	256	335
360	710	75	138	138	209	209	276	276	362
710	800	83	153	153	228	228	303	303	397
800	900	91	169	169	256	256	338	338	440
900	1000	103	188	188	280	280	365	365	480

Note: Tolerance limits for radial internal clearance in self-aligning roller bearings under no load. This table applies to bearings conforming to the Basic Plan for Boundary Dimensions of Metric Radial Bearings, AFBMA Siandards, Section 2, Table 1.

[a] These symbols relate to the AFBMA Standard Section 5 — identification code.

Strength

Cyclic contract stresses may vary from 700 to 2800 MPa (100,000 to over 400,000 psi) in operation and shock loads may range to 4100 MPa (600,000 psi). Fatigue life is inversely proportional to the 7th to 10th power of the stress under the most heavily loaded rolling element. Maximum hardnesses are limited by material sensitivity to cracking and metallurgical stability.

Due to the high-contact stress and subsurface shear stress in rolling element bearings, surface or near-surface elements must be hardened to a minimum of Rockwell C59. To support the high-subsurface stresses, the supporting core must be of equal hardness or provide a hardness gradient that insures structural integrity and fatigue life. Strength and hardness

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Table 4
APPROXIMATE TEMPERATURE LIMITS FOR ROLLING BEARING STEELS[b]

Through-hardening steel	Carburizing Steel	Temperature °C	Temperature °F
TBS-9	4118	1018	
Stroloy 503A	5120	1026	
52100[a]	8620	150	300
1070M	4620		
	4720		
	4320		
52100 Type 1	4820		
52100 Type 2	9310	175	350
440C	3310		
TBS-600	CBS-600	205	400
52CB		260	500
MHT			
14-4 (Mod. 440C)			
Halmo		315	600
M-50			
TBS-1000	CBS-1000	370	700
M-10		430	800
M-1		480	900
M-2			
WB-49		540	1000

[a] Has been used successfully at 205°C (400°F) when heat stabilized, with increased internal clearance in bearing, and very light load.
[b] Based on hot-hardness and temper resistance.

at operating temperature must be considered since commercial bearing materials are required to operate at 120°C (250°F) with no derating. At higher temperatures, either increased scatter in early life or reduced average life must be anticipated. Over 175°C (350°F), materials with high hot-hardness such as the tool steels are required. Table 4 gives commonly used materials and their temperature limits.

Where static loads are encountered with bearings not rotating, resistance to plastic indentation becomes important. Static capacities are normally chosen to insure no significant indentation of the raceways. Some bearings, however, can operate subsequently at lower loads with slight brinelling with no significant effect on system behavior. When carburized materials with higher core strength are chosen, these static loads should not exceed the crush strength of the case and it is advisable to determine these limits by laboratory tests.

Fatigue Resistance

Whereas high carbon chromium bearing alloys were once felt essential to adequate fatigue life, most materials, properly hardened, and with sufficient core strength to support the shear stress can insure satisfactory life and life dispersion. Fatigue resistance is also a function of residual stresses remaining after bearing manufacture. Tensile stresses are to be avoided and in many cases it is advisable to specially process inner rings in a manner to limit these. In those applications where abrasion, overheating, lubricant depletion, or abusive assembly stresses in the inner and outer rings are anticipated, provisions must be made to limit the stress or else reduced bearing life may be anticipated.

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Wear

Fortunately, the more commonplace materials used today, namely 52100, TBS9 (Timken®), and the various carburizing steels have excellent resistance to wear under normal operating conditions with full film lubrication. Any abrasion is detrimental due to the damaging effects of increased osculation between the rolling elements and inner rings. Wear between the rolling elements and retainer can be controlled for the most part by proper selection of the retainer materials and by optimizing the geometry of the contact of the rolling element and retainer.

Dimensional Stability

In case of thru-hardening steels, operating temperatures must be limited to 120°C (250°F) or provisions made to accommodate potential distortion/growth of the inner rings. For higher operating temperatures, parts must be "stabilized" by additional heat treatment or tempering and/or deep freezing to insure transformation of all retained austenite. In the case of carburized parts where the volume of retained austenite is small, transformation is not significant. In the transformation of retained austenite in the finished part made of conventional thru-hardening materials, shrinking is encountered after relatively few hours of operation. This is followed by growth during extended operation or exposure to higher temperature. AFBMA standards are available as a guide in selection of parts for stability.

For corrosive conditions, it is advisable to consider means for protecting the bearing other than resorting to exotic materials. Some stainless steels are used occasionally in rolling element bearing for specialized applications and must be chosen with care. Fatigue data and relative life of these materials are not well substantiated in many cases.

Retainer Materials

Commercial bearings in past years generally used a leaded bronze or an aluminum bronze for retainers. More recently, ferrous materials have been used either as strip steel or castings. Recently the use of polymeric retainers has developed, primarily heat-stabilized nylon 6-6 both with and without glass reinforcement. In ball bearings polymers are of great value since they are better able to endure the adverse operating conditions induced by moment loads.

For high-temperature applications, particularly with marginal lubrication, coated retainers generally made of various ferrous materials have been selected. The aerospace industry has utilized an iron-silicon bronze, both plated and unplated, for adequate strength at higher temperatures. Oiling systems in early gas turbine engines could supply no additional lubricant to a bearing for long periods at startup. Under these conditions, break-in or sacrificial films were essential and the bronze retainers used were generally silver plated or silver plated with a lead-tin overlay.

Nodular-graphite iron retainers have good wear properties when properly prepared and resist some gases which may be found in lubricants. In ammonia compressors, they endure the environment detrimental to most normally used bronzes. Nylon polymers are sensitive to materials with free phosphorus on the surface, to a limited number of diluents and solvents, and to some lubricants.

Breakdown products must be anticipated from operating lubricants and their additive packages at high temperatures. Material selection must anticipate attack from EP additives at high temperature. Many boundary lubricants are particularly detrimental. Moisture is doubly harmful in that it can aid breakdown products to become corrosive and attack bearing surfaces.

ROLLING BEARING THEORY

Contact of Elastic Bodies

Two basic theoretical types of contact occur between elastic solids, namely point contact

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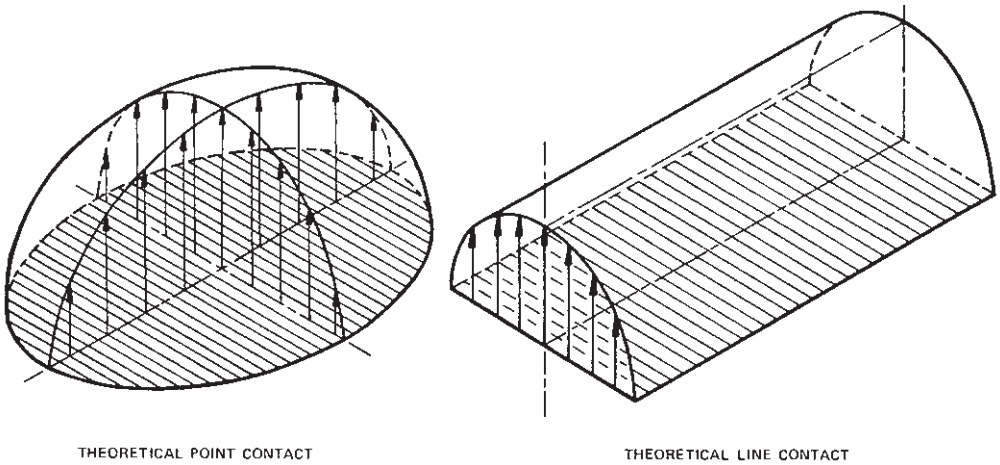

FIGURE 6. Stress distributions over elliptical "point" contact area and rectangular "line" contact (no end effects).

and line contact. Point contact refers to the conjunction of two elastic bodies such that the initial contact, under no load, is a single point. As load is applied, the bodies deform elastically and the contact spreads out into a finite area of elliptical shape. Assuming that the pressure is entirely normal to the contact surface (i.e., there are no tangential or frictional forces) the contact pressure assumes a semiellipsoidal distribution as shown in Figure 6.

As defined schematically in Figure 7, typical osculation values for ball bearings are 51 to 52% for the inner race and 52 to 53% for the outer race. For spherical roller bearings, osculation is normally given as the ratio of the radius of curvature of the rolling element to the radius of curvature of the ring raceway. Typical values range from 94 to 99%. Osculation clearance is the clearance at the end of the effective length of contact between a roller and a raceway.

Theoretical line contact is that which occurs between two cylinders with parallel axes, or a cylinder and a flat plate of infinite length in the cylinder axial direction. Under load, the bodies deform elastically and the contact area becomes a rectangle. The distribution of pressure over this contact surface is uniform in the axial direction and elliptical in the direction perpendicular to the axis, as illustrated in Figure 6. Since actual bearing contacts are not infinitely long, true line contact is only approached under certain conditions in real bearings.

Roller Crowning

If a cylinder of some finite length is loaded against another cylinder or flat, a stress concentration develops at the end of contact due to the discontinuity. For lightly loaded cylindrical or tapered roller bearings, this stress concentration is not serious. With high load or with misalignment, however, it can drastically reduce bearing life. Therefore most (but by no means all) cylindrical and tapered roller bearings utilize crowned rollers such as illustrated in Figure 8. The full crown is a circular arc that results in a contact ellipse with a length to width ratio on the order of 50. To overcome the rather short length of contact with a full crown under light load, the partial crown provides a straight length of about 50 to 60% of the roller length, with only the ends relieved. This type of crown has two disadvantages: (1) stress concentrations under heavy loads at the junction between the crown and the straight length, and (2) less tolerance of misalignment than the full crown. At least one manufacturer provides a modified full crown: flatter in the center of the roller with increasing curvature toward the ends. Typical values of roller crown radius range from 1 to

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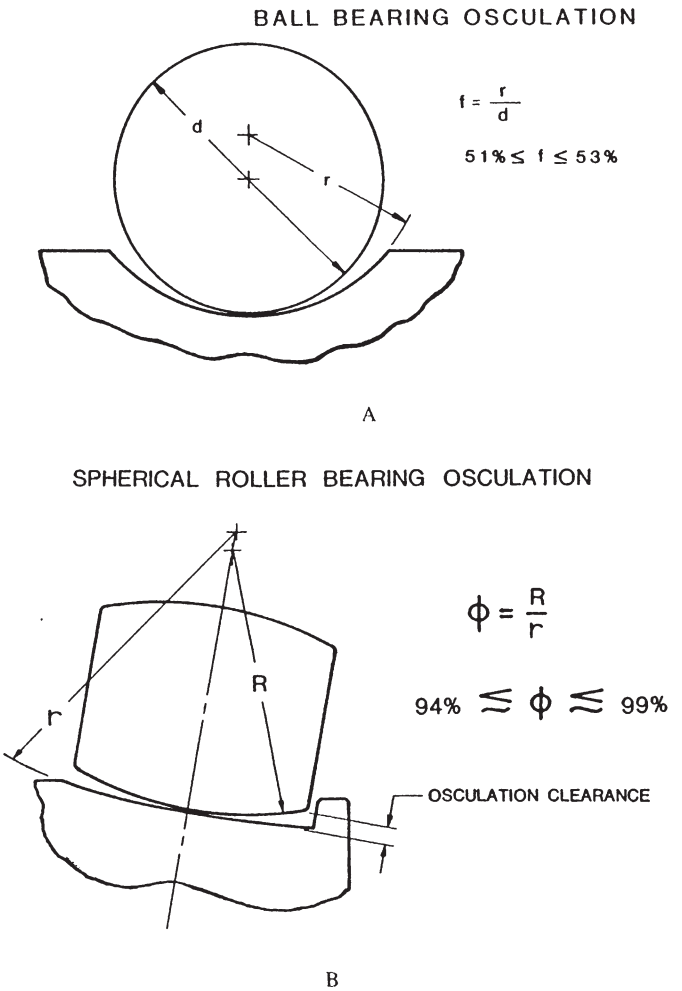

FIGURE 7. Osculation for (A) ball and (B) spherical roller bearings.

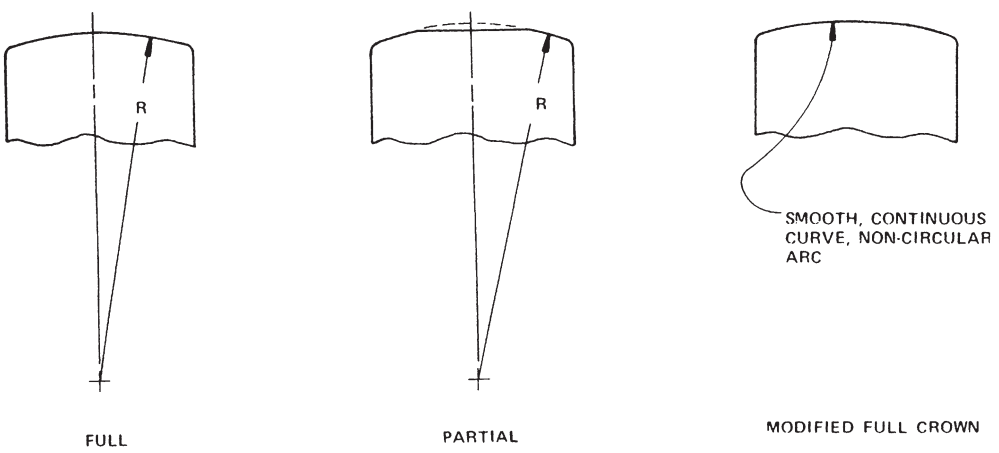

FIGURE 8. Three types of roller crowns.

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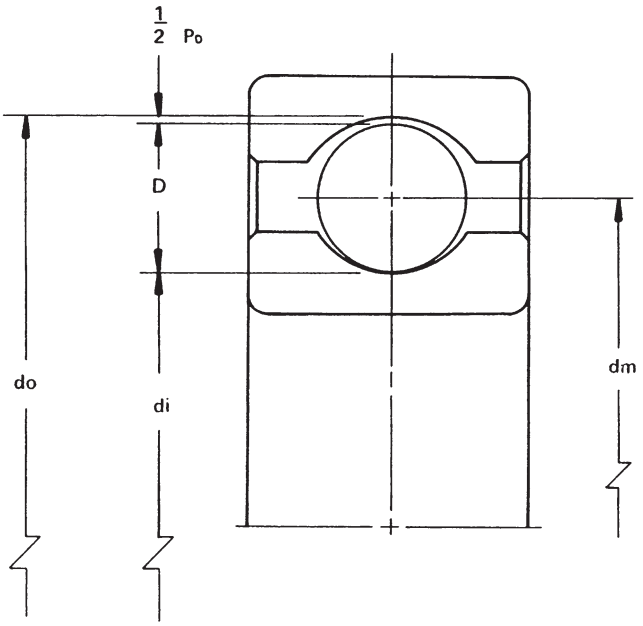

FIGURE 9. Radial ball bearing nonmenclature.

25 m (50 to 1000 in.). Osculation clearances are normally in the range of 5 to 50 μm (0.0002 to 0.002 in.).

Contact Stresses

The maximum principle stress, which is often used as a measure of load severity, is the compressive stress normal to the contacting surface at the center of the contact ellipse. Normally, however, one of the sheer stresses at some depth below the contact surface is decisive with regard to the load-carrying ability of the contact. Typically, maximum compressive stress for the most heavily loaded rolling element for normally loaded rolling bearings is in the range of 700 to 1400 MPa (100,000 to 200,000 psi), with roller bearings tending toward the lower end of that range, and ball bearings the higher. A high stress in a very heavily loaded rolling bearing would be 2500 MPa (360,000 psi). In a nonrotating bearing, stresses up to 4000 MPa (580,000 psi) and higher sometimes can be tolerated without significant impairment of subsequent operation. Reference 3 gives the most general method of calculating stresses in a concentrated contact, while References 4 to 8 give procedures particularly adapted to rolling element bearings. Hartnett[7] provides a means of estimating potential stress concentrations with crowned rollers. Hamrock[8] has developed a very simple method which may be carried out on a hand calculator.

Nomenclature

Definitions required for discussion of loads and motions within rolling bearings are listed below and are illustrated in Figures 9 to 12 for ball, cylindrical roller, spherical roller, and tapered roller bearings.

C	=	Basic load rating
D	=	Rolling element diameter
d_m	=	Pitch diameter of the rolling element set
F_a, F_r	=	Bearing applied axial load, radial load
f_c	=	A factor used to determine basic load rating (Equations 9 to 12); depends on geometry and material

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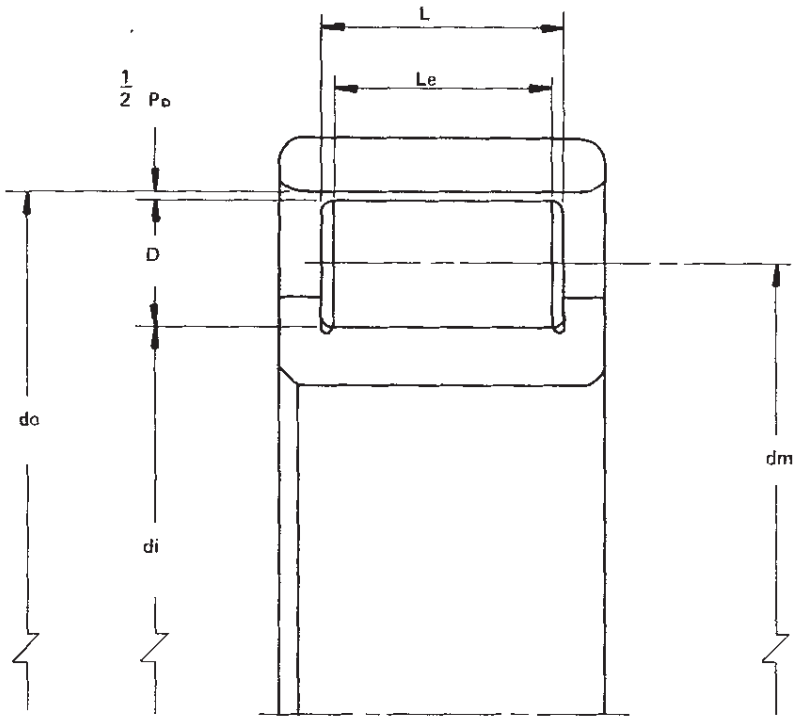

FIGURE 10. Cylindrical roller bearing nomenclature.

f_i, F_o	=	Inner and outer race osculations for ball bearing (see Figure 7)
i	=	Number of rows of rolling elements
K	=	Coefficient in the load-deflection equation; depends upon material, elastic properties, and osculation
1	=	Roller length
l_{eff}	=	Effective length of contact between roller and ring raceway
N	=	Number of stress cycles
N_1, N_o, N_c	=	Rotational speed of inner ring, outer ring, and cage
N_r	=	Rolling element rotational speed about its own axis (i.e., relative to the cage)
Q	=	Rolling element load
S	=	Probability of survival
t	=	Exponent in the load-deflection equation; depends upon osculation
V	=	Stressed volume
y	=	$D\cos\alpha/d_m$
Z	=	Number of rolling elements per row
z_o	=	Depth of maximum orthogonal shear stress
α	=	Contact angle (also 1/2 included cup angle for tapered roller bearings)
β	=	1/2 Included cone angle (tapered roller bearings)
γ	=	1/2 Included roller centerline angle (taper roller bearings)
δ	=	Deflection
λ	=	Life reduction factor for stress concentrations
ν	=	1/2 Included roller angle (tapered roller bearings)
τ_o	=	Maximum orthogonal shear stress

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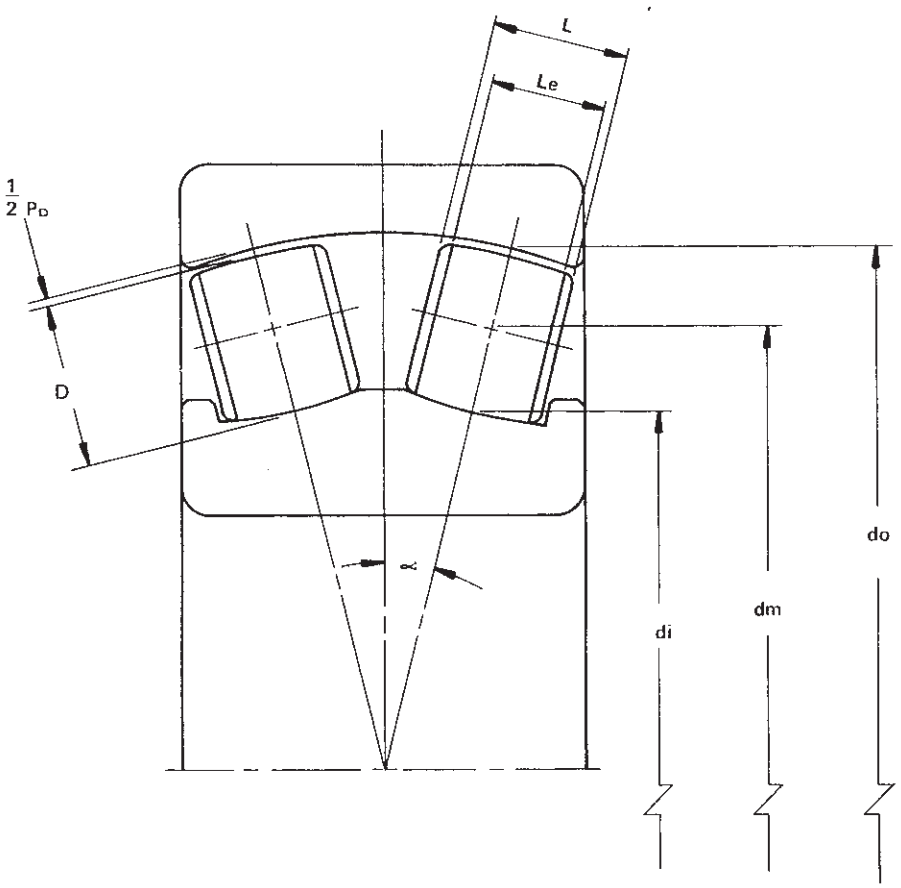

FIGURE 11. Spherical roller bearing nomenclature.

Contact Deformations and Load Distribution

To determine the magnitude of the contact stresses occurring in a rolling bearing, one must first determine the magnitude of the load on each rolling element. Figure 13 shows a free-body diagram of a radially loaded bearing. The equations of static equilibrium are not sufficient to determine the distribution of load among the rolling elements; load-deflection relationships at the rolling element-raceway contacts are also required. In addition, it is necessary to assume that the ring is round before loading and remains round after loading, and the only deflections are those at the Hertzian contacts. In general, load-deflection relationship for a Hertzian contact can be expressed by:

$$Q = K\delta^t \qquad (1)$$

Typical values for coefficient K and exponent t are given in Table 5. They do not vary greatly for rolling element contacts for bearings with the usual range of power transmission applications, say 25- to 300-mm (1- to 12-in.) shaft size. Heavily loaded ball bearings, where the contact ellipse is of significant size in relation to the ball diameter, may be somewhat stiffer than indicated by the foregoing relationship; light to normally loaded spherical roller bearings, as well as tapered and cylindrical roller bearings with crowned rollers, seldom operate with rectangular contact areas and are usually somewhat less rigid than indicated by the line contact exponent. For normal engineering purposes, however, values given in Table 5 are adequate.

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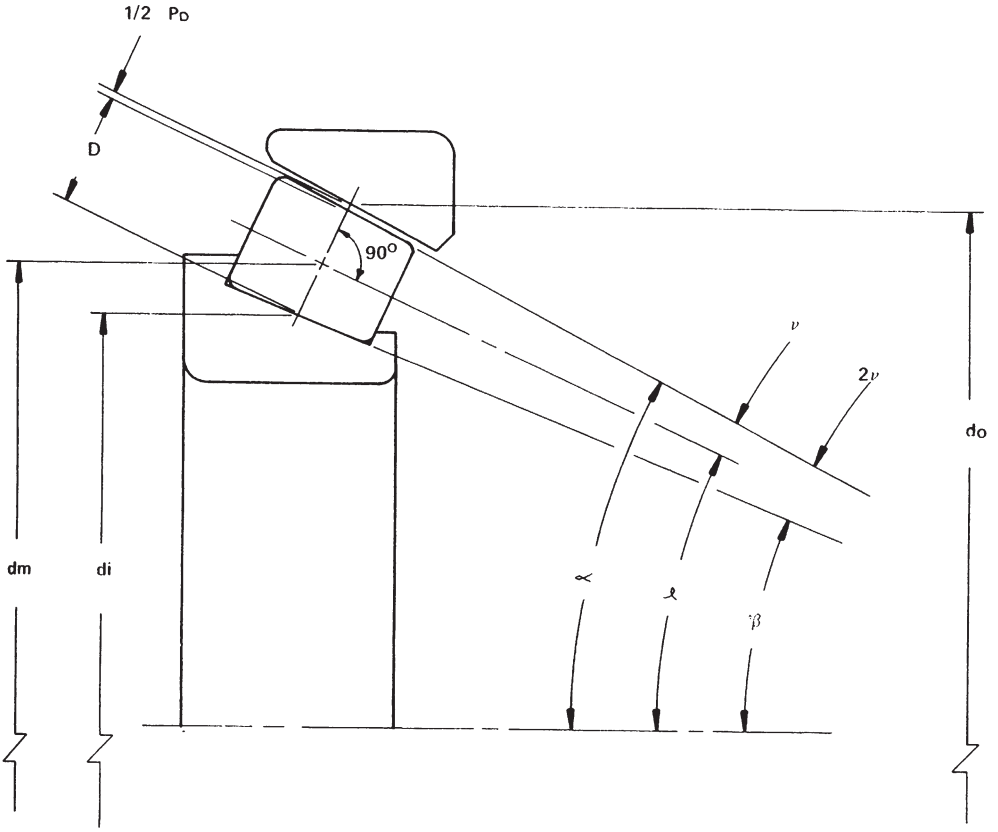

FIGURE 12. Tapered roller thrust bearing nomenclature (use cos ℓ in place of α in Equation 13.)

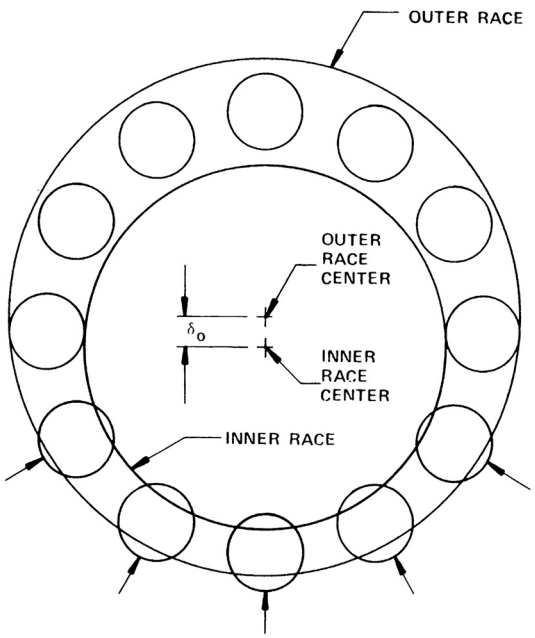

FIGURE 13. Bearing deflection under load.

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Table 5
ELASTIC CONTACTS
ORDER OF MAGNITUDE

Type of contacts	Exponent (t)	Coefficient (K)
Point	1.5	18×10^6
Line	1.11	7×10^6

Note: Load in pounds; deflection in inches.

Consider the case of an inner raceway displaced a distance δ_o with respect to the outer race as shown in Figure 13. Given the diameters of the raceways and the magnitude and direction of displacement, mutual approach of the two raceways can be calculated at any roller location. Deflection across a roller at that position is then equal to the approach less any initial clearance in the bearing. The roller load can then be calculated from Equation 1. The vector sum of all roller loads will yield the applied radial load F_r required to produce displacement δ_o. If it is assumed that the bearing has zero radial clearance and that the rings are circumferentially rigid as previously described, for any practical number of rolling elements for bearings with a load-deflection exponent of 1.5

$$Q_{max} = \frac{4.37\,F_r}{iZ\cos\alpha} \tag{2}$$

and

$$Q_{max} = \frac{4.08\,F_r}{iZ\cos\alpha} \tag{3}$$

for bearings with a load deflection exponent of 1.11. Since few bearings operate with zero clearance, the radial load on the most heavily loaded ball or roller is commonly calculated from the following equation:

$$Q_{max} = \frac{5\,F_r}{iZ\cos\alpha} \tag{4}$$

For thrust and angular contact ball and roller bearings

$$Q_{max} = \frac{F_a}{Z\sin\alpha} \tag{5}$$

Rolling Bearing Kinematics

Rotational speed of a rolling element or the cage (the orbital speed of the rolling element) can be determined by analyzing the bearing as an epicyclic gear train, if no slip is assumed at the rolling-element/raceway contacts. Significant slip is possible, however, with bearings operating under extremely light load at very high speed. For the general case where both inner and outer rings rotate, rotational speeds of the cage and rolling elements are given by the following equations:

$$N_c = 1/2[N_i(1-y) + N_o(1+y)] \tag{6}$$

$$N_r = \frac{d_m}{2D}(1 - y)(1 + y)(N_o - N_i) \tag{7}$$

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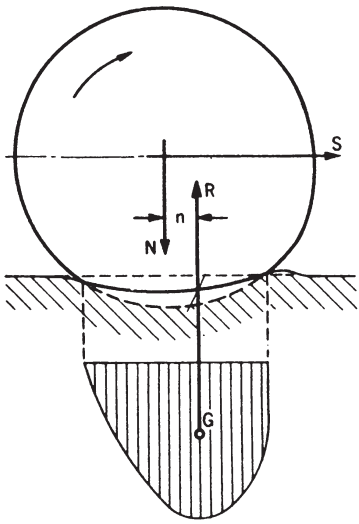

FIGURE 14. Forces on rolling element with bulge created by tangential force.

Table 6
AVERAGE BEARING COEFFICIENTS OF FRICTION MEASURED AT SHAFT SURFACE

	Coefficient of friction					
	Ball bearings		Spherical roller bearings		Cylindrical roller bearings	
Type of load	Starting	Running	Starting	Running	Starting	Running
Radial	0.0025	0.0015	0.0030	0.0018	0.0020	0.0011
Thrust	0.0060	0.0040	0.0120	0.0080	—	—

Note: Bearing torque = coefficient of friction × load × shaft radius.

For the more typical case where only one ring rotates, the speed of the stationary ring is zero in the above equations. These relationships are useful for determining centrifugal and gyroscopic forces in high-speed bearings, relating the number of contact stress cycles to inner and/or outer ring rotation, and determining the degree of slip by comparing the calculated cage speed to a measured cage speed.

Friction in Rolling Bearings

One source of resistance to rolling is the internal friction (hysteresis) due to cyclic stressing of rolling contact surfaces. Another source of friction in rolling contacts is illustrated in Figure 14. The tangential force required for rolling results in a slight bulge in front of the rolling element and a depression behind. Due to these small elastic deformations, slippage on a microscopic scale (microslip) occurs within the contact areas.

For many bearing types (such as ball bearings, spherical roller bearings, and most thrust roller bearings), sliding due simply to the lack of theoretical true rolling is another source of friction. In addition, pure sliding contacts in rolling bearings include roller-end/flange contacts, rolling-elements/cage contacts, and cage/land contacts. References 4 to 6 give methods for estimating friction torque for rolling bearings. For rough estimates of torque, Table 6 may be used.

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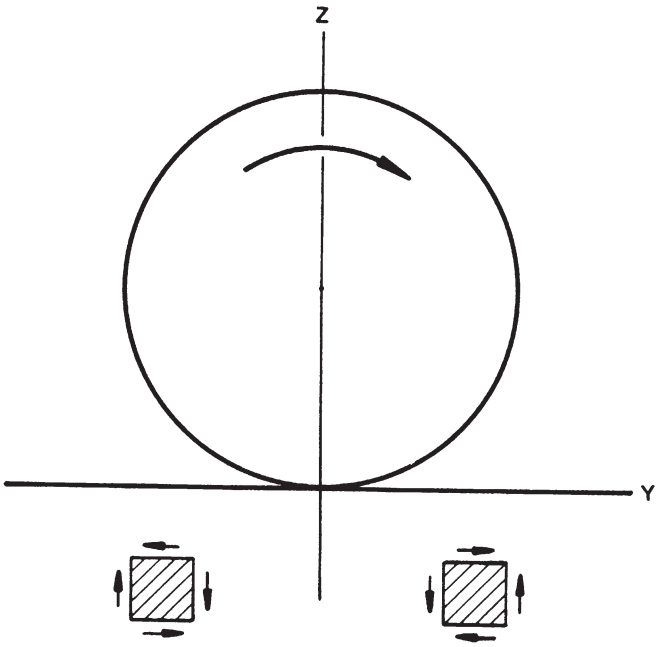

FIGURE 15. Illustration of orthogonal shear stress.

Static Capacity

Occasionally rolling element bearings are subjected to very heavy loads while stationary. To determine whether or not such loading has any effect on subsequent rotational operation, the concept of a static load rating has been developed. Permanent flat spots, or "brinelling", usually manifest themselves as noise or vibration. They do not seem to significantly effect torque or fatigue life of the bearing unless they are of extreme magnitude. Departure from elastic behavior cannot be detected at a stress level significantly less than 4000 MPa (580,000 psi). Formulas for calculation of static load ratings involve not only elastic material properties but also details of the internal bearing geometry.[10,11]

Rolling Bearing Endurance

The mathematical model generally used for rolling contact fatigue calculations was originally developed by Lundberg and Palmgren.[12,13] It may be expressed mathematically as follows:

$$\log_e\left(\frac{1}{S}\right) \propto \frac{\tau_o^c N^e V}{z_o^h} \tag{8}$$

The maximum orthogonal shear stress, τ_o, occurs on planes parallel and perpendicular to the rolling direction as illustrated in Figure 15. This stress has its maximum value at some depth below the surface and at some distance ahead of the center of the contact in the rolling direction. At that same depth and at the same distance behind the center of the contact, τ_o reaches the same magnitude but in the opposite direction. Its total amplitude determines the severity of loading for a rolling contact with regard to fatigue. Determination of τ_o involves material properties and contact geometry along with load distribution among the rolling elements. Depth z_o, also a function of these variables, is in the denominator of Equation 8 because the deeper below the contact surface that the crack initiates, the longer for it to propagate to the surface and result in a spall. The number of stress cycles, N, occurring at

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any point in the rolling contact surfaces, can be related to the number of inner ring revolutions through the kinematic relationships described earlier. The volume of stressed material, V, is included on the theory that resistance to fatigue crack initiation varies throughout the material. The greater the volume of material subjected to cyclic stress, the greater the probability of a crack initiating.

By making simplifying assumptions. Equation 8 can be rewritten in terms of bearing geometry factors. For arbitrary conditions of 90% reliability (S = 0.9) for 1 million inner ring revolutions, the following equations result.

For radial ball bearings:

$$C = f_c(i \cos \alpha)^{0.7} Z^{2/3} D^{1.8} \tag{9}$$

Note: when D > 1 in., or 25.4 mm, use exponent 1.4 instead of 1.8, and where

$$f_c = 3050 \lambda \left\{1 + \left[1.04 \left(\frac{1-y}{1+y}\right)^{1.72} \left(\frac{f_i}{f_o} \cdot \frac{2f_o - 1}{2f_i - 1}\right)^{0.41}\right]^{10/3}\right\}^{0.3} y^{0.3} \left[\frac{(1-y)^{1.39}}{(1+y)^{1/3}}\right] \left(\frac{2f_i}{2f_o - 1}\right)^{0.41} \tag{10}$$

for C in pounds and D in inches. (To obtain C in N when D is in millimeters, replace 3050 with 40.1 for D ≤ 25.4 mm and with 146.2 for D > 25.4 mm.)

For radial roller bearings:

$$C = f_c(i \cos \alpha L_{eff})^{7/9} Z^{3/4} D^{29/27} \tag{11}$$

$$f_c = 18673 \lambda \nu \left\{1 + \left[1.04 \left(\frac{1-y}{1+y}\right)^{143/108}\right]^{9/2}\right\}^{-2/9} y^{2/9} \left[\frac{(1-y)^{29/27}}{(1+y)^{1/4}}\right] \tag{12}$$

for C in pounds and l_{eff} and D in inches. To obtain C in Newtons with l_{eff} and D in millimeters, use 207.9 in place of 18673. Reduction factor λ accounts for stress concentrations within contacts and is covered in the next section.

ROLLING BEARING APPLICATION CONSIDERATIONS

In this section, an attempt will be made to point out application and operating conditions that differ significantly from the standard assumptions, and how they might affect bearing performance.

Load

Basic load rating C discussed earlier is itself an extremely high load; a load nearly as high as the C rating would not normally be applied in practice. The expected range of relative loading is indicated in Table 7, which is taken from AFBMA Standard 7. To this might be added two additional load categories: extremely light load, less than 1 to 2% of C, and extremely heavy load, greater than 25 to 30% of C.

Providing the fitting practices recommended in AFBMA Standard 7 are followed for light to heavy loading, no problems should be expected. In the extremely light load range however, especially in combination with very high-speed and high-viscosity lubricants, rolling elements sometimes slide rather than roll. This problem is not uncommon in gas turbine engine bearings. Solutions have included preloaded hollow rollers and out-of-round ring raceways.

At the other end of the scale are extremely heavy loads, usually accompanied by very low speed, such as might occur in a crawler tractor final drive. Operation is usually in the boundary lubrication regime and surface origin failures result. Also, under extremely heavy loads (i.e., above 25 to 30% of C) it is often difficult to prevent movement of a press fit

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Table 7
DEFINITION OF LOAD RANGES FOR ROLLING ELEMENT BEARINGS

Load	Ball bearings	Roller bearings
Light	Up to 7% C	Up to 8% C
Normal	From 7% C	From 8% C
	Up to 15% C	Up to 18% C
Heavy	Over 15% C	Over 18% C

inner ring on its shaft, even using the fits recommended for a heavy load. The usual solution to this problem is a threaded shaft and locknut to clamp the press fit inner ring against a shaft shoulder. These characteristics of boundary lubrication, high load, and heavy press fits combine to produce a very highly stressed component. Under such conditions it is important that the bearing and mating components (e.g., shaft, housing) are free from flaws such as nicks and dents. While rolling element bearings can be quite forgiving of such flaws under normal conditions, under extremely heavy load they can lead to premature failure and bulk cracking either as a primary or, more often, a secondary failure mode.

Load direction can often determine how well a rolling bearing performs. On angular contact bearings, axial load is simply combined with the radial load to get an equivalent radial load to estimate life. For radial cylindrical roller bearings, however, axial load is resisted only by the roller ends and ring flanges; it is not considered in the fatigue life calculation. Opposing flanges on inner and outer rings are intended to provide only axial location and not to support applied thrust loads such as occur in helical gear drives. Axial load capacity of these bearings can be improved by careful attention to such characteristics as roughness, rounding of sharp corners, roller end to race squareness, and rib to ring race squareness.

Some types of bearings and bearing arrangements (e.g., "back-to-back" angular contact bearings) are designed specifically to resist moment loading. Spherical roller bearings, on the other hand, cannot support any moment load. For other roller bearings, application of a moment load usually results in edge loading of the rollers and early failures. For ball bearings, a certain amount of moment load can be tolerated without causing the balls to ride too high in the groove. Such loading, however, is very detrimental to the typical riveted or welded ribbon-type metallic cage, and can result in early fatigue failure of the cage. Such failures are common in agricultural tillage implements subject to distortions as they bump and bounce across rugged terrain. Polymeric cages (e.g., injection-molded nylon) have largely overcome this problem.

Speed

Operating speed of a rolling bearing has ramifications other than simply determining how long it takes to reach its normal fatigue life. Very low speed can result in surface distress damage to the rolling contact due to inadequate EHD lubricant film thickness. Very high speeds can result in skidding damage, less than expected EHD film thickness due to starvation, abnormal distribution of loading between the inner and outer raceways due to centrifugal forces on rolling elements, strain on the cage due to its own centrifugal expansion and to resisting gyroscopic moments of rollers, centrifugal expansion of the gyroscopic moments of rollers, centrifugal expansion of the inner race causing a reduction or complete loss of a press fit, and centrifugal force throwing lubricant away from the inner race and resulting in ineffective lubrication. These problems generally occur only at very high speeds such as in aircraft gas turbine mainshaft bearings and solutions are reported in the literature.

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Table 8
APPROXIMATE NORMAL SPEED LIMITS FOR ROLLING ELEMENT BEARINGS

Bearing type	Normal speed limit (DN)[a]		
	Grease or oil bath	Circulating oil	Highest DN attained under ideal conditions
Radial or angular contact groove ball bearings	300K[b]	500K	3500K
Cylindrical roller bearings	300K	500K	3500K
Tapered roller bearings	150K	300K	3500K
Spherical roller bearings	150K	300K	1000K

[a] DN = bore (D), mm × speed (N), rpm.
[b] K indicates × 1000 (e.g., 300K = 300,000).

The most common problem associated with high speed, however, is simply overheating. For this reason, most bearing catalogs give a speed limit for each type of bearing. Typical values are shown in Table 8. Not absolute limits, these are simply speeds below which ordinary lubricants and lubrication methods will normally give satisfactory performance. Particular attention to bearing components, the lubricants used, and the lubricant application method enables satisfactory performance at speeds well beyond typical catalog limits. The highest speeds are also indicated for various types of bearings under laboratory conditions.

Temperature

The range of temperatures over which rolling element bearing can perform successfully depends upon the materials of the rolling bearing components, materials of auxiliary components such as seals, and lubricants.

Two aspects of material hardness-temperature relations are to be considered: short-term hot-hardness and long-term temperature resistance. The former refers to inadequate hardness of bearing steels at abnormally high operating temperature even though they recover full hardness when reduced to room temperature as illustrated in Figure 16. Temper resistance, on the other hand, refers to the ability of a bearing steel to recover its original room temperature hardness after being exposed to high temperature for some period of time as illustrated in Figure 17. Based on this phenomenon and a minimum acceptable hardness of 59 Rockwell C at operating temperatures, approximate temperature limits for bearing steels are given in Table 4.

Typical temperature limits for common rolling bearing seal materials are given in Table 9. Temperature limits for lubricants can vary considerably depending upon the additives they might contain. For example, a hypoid gear lubricant containing highly active EP additives may be limited to a significantly lower temperature than that shown in Table 10. Maximum temperatures for some of the more common nonmetallic cage materials are shown in Table 11. Metallic cages rarely limit the allowable operating temperature.

Misalignment and Contact Geometry

Even though the underlying load rating relations are based on perfect geometric shapes with smooth surfaces in perfect alignment, Lundberg and Palmgren[12,13] introduced reduction factor λ in Equations 10 and 12 to account for stress concentrations within contacts. Stress concentrations can result from, for example, edge loading due to heavy loading or improper crown, inadequate crown blend, misalignment, imperfect geometry, and rough surfaces. For ball bearings only the latter three are significant; consequently, values of λ are usually much

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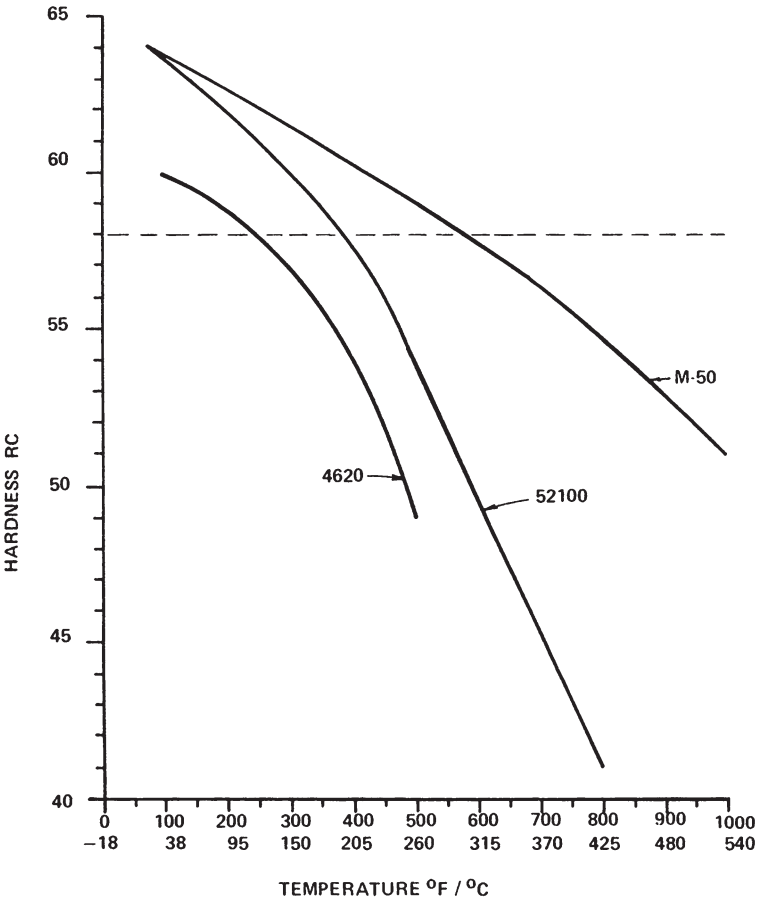

FIGURE 16. Hot hardness of representative rolling element bearing steels.

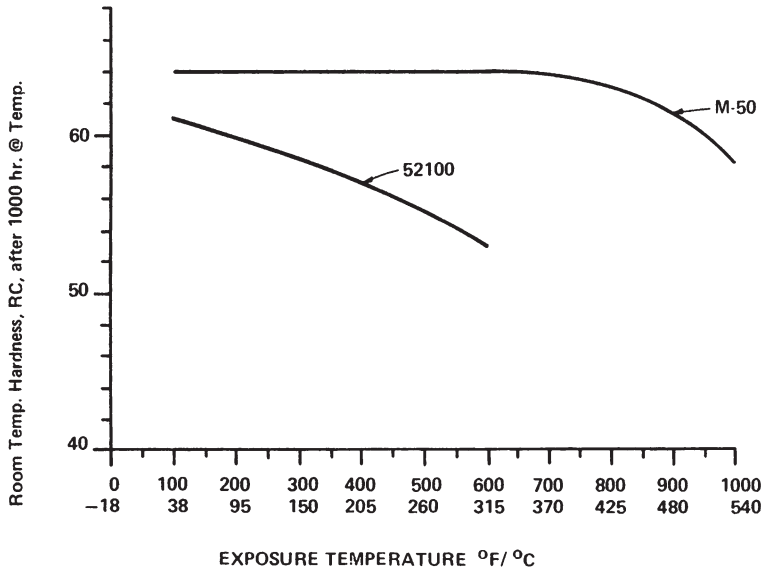

FIGURE 17. Temper resistance of representative rolling element bearing steels.

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Table 9
TEMPERATURE LIMITS FOR SEAL MATERIALS

Material	Temperature limit °C	°F
Felt	95	200
Buna N	110	225
Polyacrylic	150	300
Viton	260	500

Table 10
APPROXIMATE TEMPERATURE LIMITS FOR ROLLING BEARING LUBRICANTS

Lubricant	Temperature °C	°F
Standard petroleum greases	110	225
Special petroleum greases, mineral oils	150	300
Synthetic greases and oils	230	450
Solid film	Bearing material limit	

Table 11
TEMPERATURE LIMITS FOR NONMETALLIC CAGE MATERIALS

Material	Temperature Limit °C	°F
Heat-stabilized nylon 66 phenolic	120	250
Fiberglass-reinforced nylon 66	150	300
Polyphenylene-sulfide	230	450

higher. Values of λ in Table 12 may be used in Equations 10 and 12 and are used in the load rating standards. They are applicable to bearings made to standard tolerances associated with ABEC/RBEC Class 1 Precision. For roller bearings, this value also assumes rollers crowned so as to achieve optimized contact.

In an attempt to define good and poor alignment, Table 13 is provided. The lower value of misalignment in each range may be considered as the limit of "good" alignment. The λ factor has already provided compensation for misalignment up to this value. At misalignments greater than the higher value in the range, serious reduction in life should be expected. Between these two values, performance will depend strongly upon the specific design as well as load, speed, and other operating conditions. Some bearings may perform satisfactorily, others may suffer premature failure.

Clearance and Ring Support

Internal bearing clearance is usually defined as the total possible radial play of one ring

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Table 12
LIFE REDUCTION FACTOR FOR STRESS CONCENTRATIONS

Bearing type	λ
Ball bearings	
Single-row radial	
Double row radial	0.95
Angular contact	
Ball bearings	
Double row radial	0.90
Contact	
Ball bearings	
Self-aligning	1.0
Roller bearings	
Cylindrical, tapered, and spherical, crowned rollers	0.61[a]

[a] Use in conjunction with $\nu = 1.36$ so that $\lambda \nu = 0.83$.

Table 13
ALLOWABLE MISALIGNMENT WITH NO REDUCTION IN LIFE FOR ROLLING BEARINGS UNDER NORMAL LOAD

Bearing type	Misalignment
Cylindrical and tapered roller bearings	0.0005—0.001 in/in.
Ball bearings	0.003—0.005 in/in.
Spherical roller bearings	0.008—0.025 in/in.

with respect to the other with no external load. For some bearing types, axial clearance, or end play, is more significant. Radial clearance can be difficult to measure and the definition is not especially meaningful for some single row, angular contact bearing types (such as tapered roller bearings) where the radial play depends upon the relative axial location of the two rings. A more rigorous and general definition can be expressed as follows in terms of component dimensions:

$$P_D = (d_o - d_i)/\cos\alpha - 2D \tag{13}$$

If the calculated clearance P_D is positive, it is called clearance. If negative, i.e., if the roller is larger than the available space, it is called preload. Some bearing types, principally tapered roller bearings, are adjustable: the user sets the clearance or preload when mounting the bearing. For such bearings clearance is usually determined by measuring end play, while preload is determined by measuring bearing rotational torque. Some angular contact ball bearings, while theoretically adjustable, are actually made to be mounted in a manner that precisely determines the internal clearance. For the many nonadjustable bearing types, such as single row deep groove ball bearings, cylindrical roller bearings, and spherical roller bearings, the initial unmounted clearance is built into the bearing by selective assembly by the bearing manufacturer. These are the clearances that are standardized for various bearing types and sizes.

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When a bearing is installed, at least one ring (sometimes both) is installed with an interference fit. This results in a change in the ring diameter of from 50 to 80% of more of the amount of interference. This reduced clearance may be called clearance. Once a bearing has reached its equilibrium operating temperature, clearance may be further changed by thermal expansion or contraction. For most applications operating in a typical ambient temperature with no external heat sources, temperature effects may be safely neglected.

Once a bearing is mounted and subjected to radial load, one ring is displaced eccentrically with respect to the other and clearance is described by angular extent of the arc wherein the rolling elements are loaded. A radial bearing with precisely zero operating clearance will have a "load zone" of 180°. For the normal case of some nominal clearance, the load zone can vary significantly as radial load varies from light to heavy. For zero clearance or a heavy preload, the load zone may not appreciably change from either 180° or 360°, respectively, for a fairly wide range of loads. For the case of a light preload and heavy applied radial load, the load zone may vary significantly between limits of 180 and 360°. Finally, the load zone can vary greatly for angular contact bearings with varying ratio of radial to axial load.

The theoretical derivations used by Lundberg and Palmgren[12,13] assumed that radially loaded bearings have precisely zero operating clearance. However, the formulas contain several empirical coefficients and exponents that were determined from fatigue life tests of actual bearings that contained some clearance. Consequently the methods given are applicable to rolling element bearings containing some nominal clearance.

If the bearing is assembled in accordance with the manufacturer's standard unmounted clearance specification, is mounted with shaft and housing fits corresponding to normal loading, and operates within a temperature range acceptable for standard bearing components, one need not be concerned about internal clearance. Usually when a bearing is assembled to some nonstandard unmounted clearance range, it is because of an unusual loading condition or expected unusual temperature condition and not because some nonstandard operating clearance is sought.

Because of the greater resiliency of the contact, ball bearings are commonly assembled with much less clearance than roller bearings. A slight preload under the extremes of normal clearance and fitting practice need not be cause for concern; a light preload can yield greater than "standard" life due to the better distribution of load. Roller bearings, on the other hand, are usually much stiffer and require more clearance; preload may not be tolerated. The bearing manufacturer should be consulted for operation outside the following limits:

Ball bearings:	Preload greater than 5 to 10 μm
	A load zone much less than about 160°
Roller bearings:	A load zone greater than 180°
	A load zone much less than about 120°

These upper limits apply of course only to radial contact bearings, or both rows of double row angular contact bearings under radial load. Single-row angular contact bearings, including thrust bearings, operate quite normally with load zones of 360°.

The preceding discussion of clearance assumes that both inner and outer rings are adequately supported. This is a satisfactory assumption for the great majority of commercial bearing applications. Some notable exceptions however are planet gear bearings, such as illustrated in Figure 18, and metalworking backup roll bearings. Calculation of bearing life for such applications is beyond the scope of this chapter.

Mounting

Most bearing manufacturers' catalogs or service instructions give the information necessary to accomplish a successful mounting. In addition, AFBMA publishes standardized shaft and housing fits, along with a method for selecting the appropriate fits for most applications.

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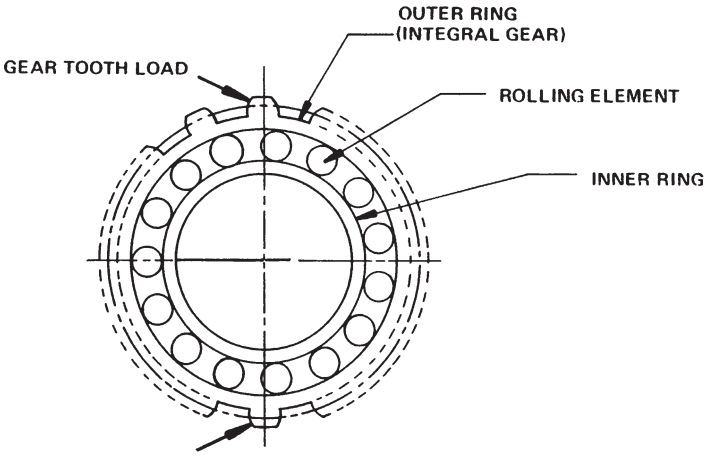

FIGURE 18. Planet gear bearing showing gear tooth loading.

The mounting arrangement must be stiff enough to support all the static and dynamic, radial and axial loads. In addition, the mounting often must provide resistance to axial movement of the rings with respect to the shaft or housing. Finally, if excessive wear is to be avoided, circumferential motion of the rings with respect to the shaft or housing must be prevented. As a general rule, the ring that rotates with respect to the load must be secured against circumferential movement relative to its supporting member. Excessive clearance at the shaft or housing bearing interface can result in either ring changing shape under load to concentrate the load on too few rolling elements. Likewise, abrupt changes in housing shape, or webs for stiffening, can result in "hard" spots in outer ring support.

Normally the most reliable means of mounting an inner ring on a shaft is with an interference fit. Specific dimensions of the fit should be established in accordance with AFBMA standards and the fit is usually accomplished either by pushing the inner ring squarely onto the shaft or by heating the inner ring so that it may be slipped onto the shaft and then allowed to shrink tight. When the shrink-fit method is used, the inner ring should be heated in an oil bath or a hot air oven (far removed from the heating elements in either case) to a temperature not greater than about 120°C (250°F) to avoid metallurgical transformations. Other possible causes of unsatisfactory interference fits include a low shaft shoulder, out-of-roundness, rough surfaces, concentricity or angularity errors in stepped shafts, and excessive shaft deflection.

Another popular inner ring mounting is the split sleeve adapter with a tapered OD, locknut, and lockwasher, used in conjunction with a tapered bore inner ring. The locknut on the sleeve is tightened until a measured amount of internal clearance is removed from the bearing. It is a reliable mounting style and allows the use of much looser shaft tolerance. Cost of manual labor in tightening and adjusting clearance usually more than offsets savings in cost of the shaft, however, and tightening to a precise axial position is difficult.

The inner ring of many small size (up to 100 mm shaft) bearings are locked to the shaft with either setscrews or a cam locking collar. Setscrews must be adequately tightened for successful performance and must either be at least 15 points Rockwell C harder than the shaft, or the shaft must be spot drilled or flatted. The eccentric locking collar, locked in the direction of shaft rotation, relies upon eccentric pinch for its considerable holding power. All of these mounting types are relatively inexpensive and easy to install with no precise adjustment. They do require, however, a very close clearance fit between the inner ring bore and shaft with rather close shaft tolerances. They also are limited in their load carrying ability. The capacity of a slip fit mounting is given by Price and Galambus.[15]

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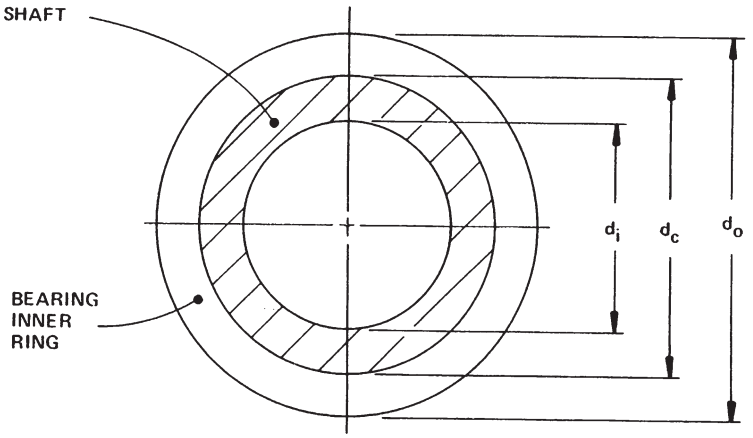

FIGURE 19. Inner ring/shaft nomenclature. Symbols: P_c = constant pressure at interface, δ = effeclive interference, E_i = modulus of elasticity (inner membrane), E_o = modulus of elasticity (outer membrane) ν_i = Poisson's ratio (inner membrane), ν_o = Poisson's ratio (outer membrane), and Δd = change in diameter. Any consistent system of units may be used.

Housing, or outer ring mountings, encounter many of the same problems. Housings do perform some other functions such as providing a lubricant reservoir and providing seals to prevent the egress of lubricant and ingress of contaminants. Housings are often purchased from the bearing manufacturer, may be either solid or split, and are usually sized for a loose fit with the bearing OD. Where the load is stationary with respect to the inner ring and rotates with respect to the outer ring, however, the outer ring is usually mounted with an interference fit. An example would be a vibrating screen.

Both shaft and housing mounting are susceptible to fretting. This physical, chemical wear phenomenon results from relative motion of very small amplitude between surfaces. Fretting is especially troublesome in the slip fit mounting, and determines the limiting load given above.[15] It also occurs in other mounting types if the interface pressure at mounting surfaces is insufficient to support the shear stresses developed by application loads.

For many types of angular contact bearings, such as tapered roller bearings, the mounting arrangement must provide for adjustment of clearance within the bearings. The change in diameter of a press fit ring can be estimated from the following equations (see Figure 19 for definitions of terms):

$$P_c = \frac{\delta}{\frac{d_c^2 + d_i^2}{E_i(d_c^2 - d_i^2)} + \frac{d_o^2 + d_c^2}{E_o(d_o^2 - d_c^2)} - \frac{\nu_i}{E_i} + \frac{\nu_o}{E_o}} \tag{14}$$

$$\Delta d_o = P_c \left(\frac{d_o}{E_o}\right)\left(\frac{d_c^2}{d_o^2 - d_c^2}\right)$$

$$\Delta d_i = -P_c \left(\frac{d_i}{E_i}\right)\left(\frac{d_c^2}{d_c^2 - d_i^2}\right) \tag{15}$$

Where both members are steel and $d_i = 0$, e.g., an inner ring on a solid shaft, the above equations reduce to

$$\Delta d_o = \left(\frac{d_o}{d_c}\right)\delta \tag{16}$$

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Materials and Processing

For cylindrical and needle roller bearings, one or both rings are often omitted with the rollers running directly on the shaft or in the housing. To achieve "catalog" life, shaft or housing material must meet the same hardness and cleanliness standards as the omitted ring. Where a lower hardness than R_c59 must be used, the reduction in life can be estimated from the following:

$$\frac{L_{10} \text{ @ hardness H}}{L_{10} \text{ @ } 59R_c} = e^{0.1(H-69)} \quad (17)$$

Cleanliness for bearing steels is usually specified in accordance with ASTM Specification A295 for through hardening steels or A534 for carburizing steels. Both of these specifications require vacuum degassing to minimize nonmetallic inclusions. Premium vacuum remelted steels with extraordinarily few and small nonmetallics are used for special high-reliability bearings such as for aircraft gas turbine engines with three to five times standard life.

ROLLING BEARING LUBRICATION

Basic Principles

The principle function of a lubricant in a rolling element bearing is to minimize friction and wear. Other important functions may include (1) protection from corrosion, (2) dissipation of heat, (3) exclusion of contaminants, and (4) flushing away of wear products. It has long been recognized that bearings can operate for extended periods under certain conditions with no evidence of wear. Presence of the original grinding, honing, or other finishing marks suggested that an oil film separated the rolling surfaces. It wasn't until Grubin and Vinogradova[16] combined elasticity and pressure viscosity relationships with hydrodynamic theory, however, that an acceptable explanation was obtained for the mechanism of forming an oil film. Dowson and Higginson[17] subsequently developed solution techniques that do not require the prior assumption of a Hertzian pressure profile. Details of elastohydrodynamic theory and related calculations are provided in an earlier chapter.

By applying EHL calculations for the conditions shown schematically in Figure 20, film thicknesses are readily determined to be of the same order of magnitude as surface roughnesses. These roughness values for rolling bearing components commonly range from a low of 1 to 2 μin. AA for balls or rollers to a high of 16 μin. AA or more for as-ground raceways.[18] It is reasonable to define the following three regimes of lubrication.

Full film — In this regime, bearing geometry, application conditions, and lubricant properties combine to yield a lubricant film so thick that there is no contact of even the highest peaks through the film. Unusually long bearing lives can be obtained.

Boundary lubrication — Here the lubricant film is practically nonexistent and the load is supported by intimate contact of the rolling component surfaces. Under these conditions bearings life is not readily predictable and often is rather short. Physical and chemical properties of the system components (e.g., bearing, lubricant, environment) all play a role in determining overall performance.[19-22]

Partial film — In this transition regime the load is supported partially by an EHL film and partially by intimate contact of the rolling surfaces. If the film is so thin as to allow significant interaction of asperities, performance may be more nearly like boundary lubrication. On the other hand, if the film is thicker with only occasional interaction of the highest peaks, then the contact may behave more nearly as full film. It is within this regime that most bearings operate and can be expected to achieve "catalog" life.

Specific film thickness Λ, the ratio of EHD film thickness to composite surface roughness, has come into use to define the state of lubrication. Composite surface roughness, in turn,

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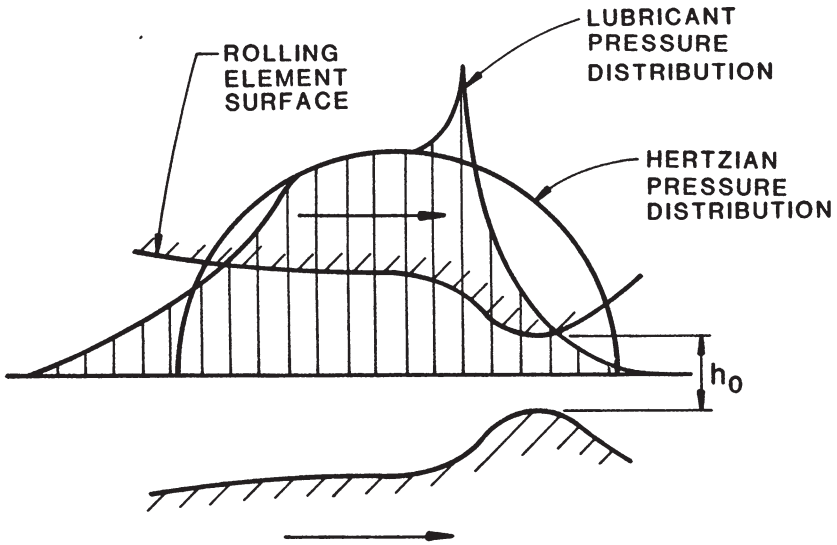

FIGURE 20. Pressure distribution in EHD contact with Newtonian fluid.

may be taken as the sum of the AA (CLA) roughnesses of the two rolling contact surfaces. For Λ greater than about four, full EHL film conditions arc obtained; for Λ values less than about one, boundary lubrication conditions apply; and for Λ between one and four, partial film conditions exist. Correlations of the Λ value with a fatigue life adjustment factor (see previous discussion on loading ratings) are shown in Figures 21 and 22.

A summary of EHL formulas applicable to rolling element bearings is given in Table 14. Two notes of caution are in order. First, operation at a lambda value less than one may result in unpredictable performance. Figure 22 for example, is strictly applicable only to the specific combination of bearing materials, lubricants, and environments used to derive the graph. A second note of warning has to do with high-speed operation. Even though a bearing may be supplied with copious amounts of lubricant, the lubricant can be pushed out of the rolling element path and not have sufficient time to flow back before the next rolling element passes that point.[26] Under such conditions, the lubricant film thickness will be less than expected. Although there is no general rule of thumb, this could occur at speed as low as 200,000 to 300,00 DN. For grease lubrication, film thickness may only be on the order of 50 to 75% of that estimated from base oil properties.

Mineral Oil Lubricants

Greases

Grease is a combination of petroleum or synthetic oil and 3 to 30% or more of a suitable thickener. Grease consistency or stiffness is determined primarily by the thickener percentage and the base oil viscosity. A grease of given consistency may be compounded in many ways by varying thickener percentage and oil viscosity. Because of this, greases of equal stiffness cannot be assumed equal in performance.

In the past, the most widely used thickeners were either sodium or calcium soaps; modern greases, however, contain many types of thickeners. Each grease has certain characteristics, but categorizing them for usage according to thickener type can be misleading. For general industrial use, sodium, calcium, lithium, aluminum, and mixed soap types, and various synthetic and nonsoap-base greases are widely used. Considerable effort has been directed to development of a multipurpose lubricant, but no grease yet satisfies all requirements.

Grease will deteriorate as a result of time, temperature, shear (mechanical working in the bearing), and contamination. These characteristics are therefore measured by standard tests

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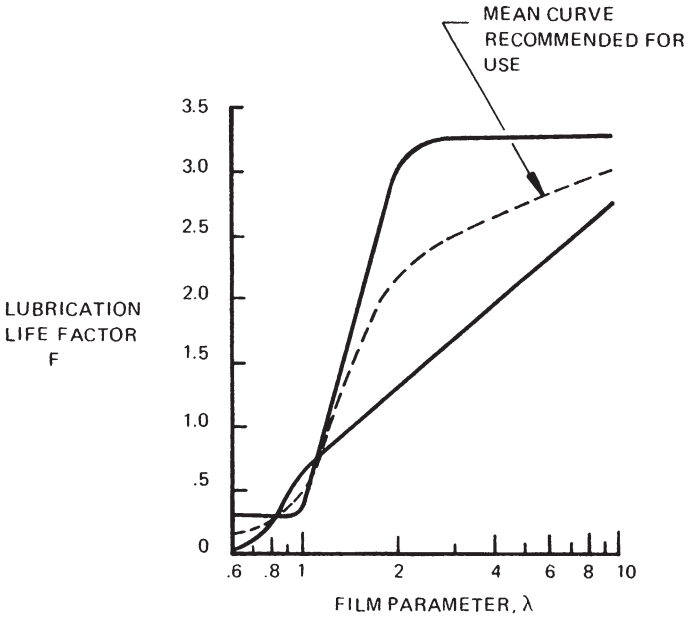

FIGURE 21. Lubrication life adjustment factor for ball and roller bearings. (From *Life Adjustment Factors for Ball and Roller Bearings, An Engineering Design Guide,* American Society of Mechanical Engineers, New York, 1971. With permission.)

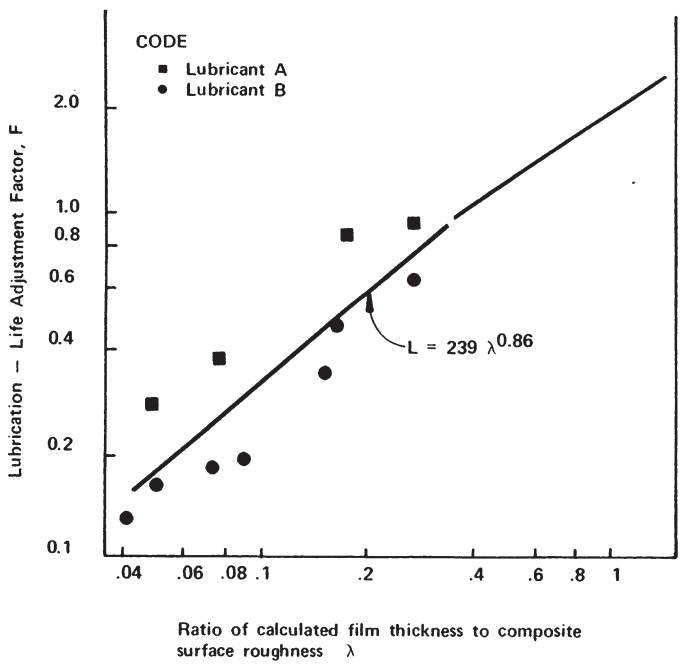

FIGURE 22. Lubrication life adjustment factor for tapered roller bearing. (From Danner, C. H., *ASLE Trans.,* 13, 241, 1970. With permission.)

of oxidation resistance, water resistance, mechanical stability, oil separation, dropping or melting point, and evaporation. Stiffness is measured by a hardness or penetration test. A change in this property during operation may have various causes: excessive working or

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Table 14
EHD FILM THICKNESS FORMULAS

For Point Contacts[25]

$$H = 3.63\,U^{0.68}\,G^{0.49}\,W_p^{-0.073}\,[1 - e^{-0.68k}]$$

For Lin Contacts [17]

$$H = 1.6\,U^{0.7}\,G^{0.6}\,W_l^{-0.13}$$

Where

$$H = \frac{h_o}{R}$$

h_o = Minimum film thickness

$$R = \frac{1}{1/R_1 + 1/R_2}$$

R_1, R_2 = Radii of curvature of body 1 and body 2, respectively, in plane parallel to rolling direction

$$U = \frac{\eta_o\, u}{E'\, R}$$

η_o = Absolute viscosity at contact entry conditions

u = Mean surface velocity = $(u_1 + u_2)/2$

$$E' = \frac{2}{\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}}$$

ν_1, ν_2 = Poisson's ratio for body 1 and body 2, respectively

E_1, E_2 = Young's modulus for body 1 and 2, respectively

$G = \alpha E'$

α = Lubricant pressure-viscosity coefficient

$$W_p = \frac{F}{E'\, R^2}$$

F = Applied load

$$W_l = \frac{p}{E'\, R}$$

p = Load per unit length of contact

Note: Any consistent system of units may be used.

churning, oil separation or vaporization, change in oil viscosity due to oxidation, etc. In some cases greases have been found excellent in laboratory evaluation and completely unsuitable in field performance. In other instances the reverse has been true. For this reason,

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field testing and field development of lubrication requirements for a particular equipment installation are often necessary.

Bearings and bearing units are designed for service ranging from nonregreasable (lubricated-for-life) to almost continuous relubrication by means of automatic systems. Advantages of grease over oil lubrication include the ease of sealing it within the bearing, the ability of grease to seal out contaminants, and its ability to coat parts and provide good corrosion protection. Disadvantages of grease include its inability to remove heat or flush away wear products, the possibility of accumulating dirt or other abrasive contamination, and a potential incompatibility problem if thickeners of different types are mixed.

Oils

Oil can be pumped, circulated, filtered, cleaned, heated, cooled, and atomized. Its advantages over grease include its ability to remove heal, flush away wear products and contaminants, and to be recycled. It is more versatile than grease and is suitable for many severe applications involving extreme speeds and high temperatures. On the other hand, it is more difficult to seal or retain in bearings and housings. Oil level or oil flow in high-speed bearings is critical and must be properly controlled.

Selection of proper oil viscosity is essential and is based primarily on expected operating temperature, speed, and bearing geometry. Excessive oil viscosity many cause skidding of rolling elements and undue lubricant friction with severe overheating and raceway damage. Insufficient oil viscosity may result in metal contact and possible premature failure. Other oil properties such as viscosity index, flash point, pour point, neutralization number, carbon residue, and corrosion protection are of varying significance in specific installations.

Synthetic Lubricants

Development of synthetic lubricants was initially prompted largely by the extreme environmental demands of military and aerospace activities. Currently the following classes of synthetic oils are available as bearing lubricants: (1) synthetic hydrocarbons, such as alkylated aromatics and olefin oligomers, (2) organic esters, such as dibasic acid esters, polyol esters and polyesters, (3) others, such as halogenated hydrocarbons, phosphate esters, polyglycol ethers, polyphenyl ethers, silicate esters, and silicones, and (4) blends, which would include mixtures of any of the above.

Use of a synthetic lubricant in a commercial application may be dictated by extreme operating conditions, for fire resistance, to meet a specification or code requirement, or to conserve petroleum-based lubricants. Although synthetic lubricants usually permit a much broader operating temperature range, temperature limits for synthetics are often misunderstood. For example, in various aircraft and space applications operation at extremely high temperature is essential but life requirements may be very short. Since industrial requirements are usually for much longer periods of operation, temperature limits for a given synthetic in industry can be much lower. Some synthetic lubricants may also have other limiting characteristics such as in load-carrying ability and high-speed operation.

Dry Lubricants

Dry, or solid lubricants are usually used under conditions of high temperature or where boundary lubrication prevails. For example, notable success has been achieved by solid lubrication of kiln car wheels, conveyor wheels, and furnace roll bearings. These high-temperature applications involve extremely low speed where ample torque is available to rotate the bearing at a relatively high coefficient of friction.

Solid lubricants may simply be dusted as a dry powder on parts to be lubricated, or they may be placed in a liquid carrier. The liquid may either be a fluid intended to evaporate or it may itself be a lubricating liquid or grease. Solid films are also applied as a bonded

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Table 15
TYPICAL OPERATING TEMPERATURE LIMITS FOR GREASES

Thickener	Temperature limit °C	Temperature limit °F
Calcium soap	65	150
Calcium complex		
Sodium soap	120	250
Sodium-calcium		
Lithium soap		
Aluminum complex	150	300
Synthetics	205	400

coating. Some of the more common materials used are graphites, molybdenum disulfide, cadmium iodide, and fluorinated polyethylenes. Typical bonding agents are resins, silicone, ceramics, and sodium silicate. Another method incorporates the lubricant into one or more of the bearing components, typically a bearing retainer. Soft metals such as silver and tin could be used for this process. In such cases the dry lubricant is transferred from the cage to the bearing raceways by the rolling elements rubbing against the cage. Bearing life is governed by the wear-out life or depletion of the lubricant. Since these special bearings are usually quite expensive, practical industrial practice is to design equipment for use of conventionally lubricated bearings.

Lubricant Temperature Limits

Temperature is the major factor affecting life of a rolling bearing lubricant. Lubricant temperature is influenced primarily by bearing speed, bearing load, ambient temperature, and lubricant system design. With two different greases used on identical applications, base oil type and viscosity, thickeners, and chemical structure can all contribute to different operating temperatures. Some greases will churn in high-speed bearings and cause overheating, whereas a channeling type grease may function satisfactorily at a much reduced temperature.

Extremely low temperatures must also be considered. The lubricant must permit an acceptable starting torque and must not freeze or become too stiff. While the lubricant must permit equipment turnover at the lowest temperature, it must also have adequate viscosity at the higher operating temperatures to provide sufficient oil film strength. For example, a petroleum type lubricant with very low viscosity oil considered for startup at –40°C and operation at 40°C may be unsuitable for operation at 80°C. In such cases, a synthetic oil or grease may be required to function satisfactorily at both the high and low limits.

Tables 15 and 16 give approximate operating temperature limits for greases and oils. As mentioned previously, however, performance can vary widely depending upon the specific details of a given application. Additives can also affect the suitable operating temperature limits. They can, for example, be somewhat extended by oxidation-inhibiting additives or they may be somewhat reduced by EP or antiwear additives. Earlier chapters of this handbook, along with References 27 through 29, provide more detailed information on various lubricant factors. Consultation with a reputable lubricant supplier is highly recommended.

Lubricant Selection

Table 17 illustrates "critical" ranges of extreme load, speed, or temperature where special

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Table 16
APPROXIMATE OPERATING TEMPERATURE LIMITS FOR OILS

Lubricant	Long term[a] °C	Long term[a] °F	Short term[b] °C	Short term[b] °F
Petroleum oils	95—120	200—250	135—150	275—300
Superrefined petroleum oils	175—230	350—450	315—345	600—650
Synthetic hydrocarbons	175—230	350—450	315—345	600—650
Organic esters	175—190	350—375	220—230	425—450
Phosphate esters	95—175	200—350	135—230	275—450
Polyglycols	160—175	325—350	205—220	400—425
Polyphenyl ethers	315—370	600—700	425—480	800—900
Silicate esters	190—220	375—425	260—290	500—550
Silicones	220—275	425—525	315—345	600—650
EP oils	60—80	140—175	65—85	150—185

[a] Long term ≈ hundreds of hours.
[b] Short term ≈ few hours.

From Szeri, A. Z., *Tribology — Friction, Lubrication, and Wear,* Hemisphere Publ., Washington, D.C., 1980. With permission.

lubrication provisions may be required. Whether an application is "normal" or critical", optimum performance is obtained when lubricant viscosity is selected for full film separation of the metal surfaces. Rotational speed is by far the most significant factor. While the required viscosity may be determined by calculations involving all pertinent variables according to the equations summarized in Table 14, a good approximation for standard bearings and a normal range of conditions may be found from the following:

$$\text{viscosity at operating temp., SUS} = 70 \left(\frac{\text{maximum catalog speed}}{\text{actual speed}} \right)^{0.7} \quad (18)$$

In this approximation, the recommended viscosity for all bearings at their maximum normal speed limit and lubricated with a naphthenic type oil is a constant 70 SUS at operating temperature. This relationship is considered valid up to the normal speed limit for a given bearing except for some adjustments in the heavy-load or low-speed range as indicated in Table 17. Some modificiation of the viscosity calculation may be necessary with synthetic lubricants due to differences in the pressure-viscosity coefficient.

Table 17 provides a general guide to lubricant selection for bearings operating in the "critical" range. In this guide many other environmental factors and maintenance procedures are presumed to be normal. Moisture, dirt, radiation, chemicals, vacuum, and oxidizing atmospheres may also influence lubricant selection. Consultation with the bearing manufacturer or a lubricant supplier is recommended when unusual conditions exist.

Lubricating Systems

A large variety of manual lubricating devices are available, including drip feed from cups. These devices permit longer intervals between manual servicing, but still require trained personnel and rigid observance of servicing schedules. A lubrication system can improve reliability. There are three basic types of centralized lubrication systems: centralized greasing, centralized circulating oil, and oil mist or fog systems. These systems are often custom-designed for individual application; their installation should be considered in the initial planning of any machine or machinery complex.

Grease lubrication systems generally may be adjusted for amount and frequency of lu-

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Table 17
CRITICAL OPERATING RANGES FOR ROLLING ELEMENT BEARING LUBRICATION

Critical range	Oils	Greases	Synthetics (fluid or grease)
Load—high Over 18% of bearing "C" rating or shock and vibration	Increase calculated viscosity by factory of 1.5 to 2.0 EP types desirable	Extreme pressure types recommended; viscosity of oil in grease in chosen on same basis as for oil lubrication	Use with caution as some products may not provide adequate film strength; consultation recommended
Speed—high Over 75% of maximum cataloged speed	Oils may be subject to foaming, and high temperatures, and should be selected carefully; circulation or frequent replacement may be required; oil levels must be controlled	Grease not normally recommended; some products suitable depending on other operating conditions; consultation advisable	Use with caution; consultation recommended
Speed — low Up to 10% of maximum cataloged speed	Use EP type of highest viscosity possible; computed viscosity may be increased by factor of 2.0	Use EP types having highest viscosity oil content; prefer softer grade than used on medium-speed applications	High viscosity and EP characteristics desirable; consultation recommended
Temperature — high Over 95°C (200°F)	Deterioration rate increases rapidly above 95°C (200°F); circulation or frequent replacement indicated; select high-quality oils with low-carbonizing characteristics	Deterioration rate increases rapidly above 95°C (200°F); regreasing frequency must be increased accordingly and must often be determined experimentally	Superior in heat resistance and operating temperature range; prefer medium speeds and medium or light loads
Temperature — low Under −5°C (20°F)	Requires oil viscosity to permit cold start and correct viscosity at subsequent operating temperature; many conventional oils suitable to −34°C (−30°F)	Requires grease to permit cold start and not thin excessively at subsequent operating temperature; many conventional greases suitable to −29°C (−20°F)	Usually required at temperatures lower than −29°C (−20°F); excellent temperature range characteristics

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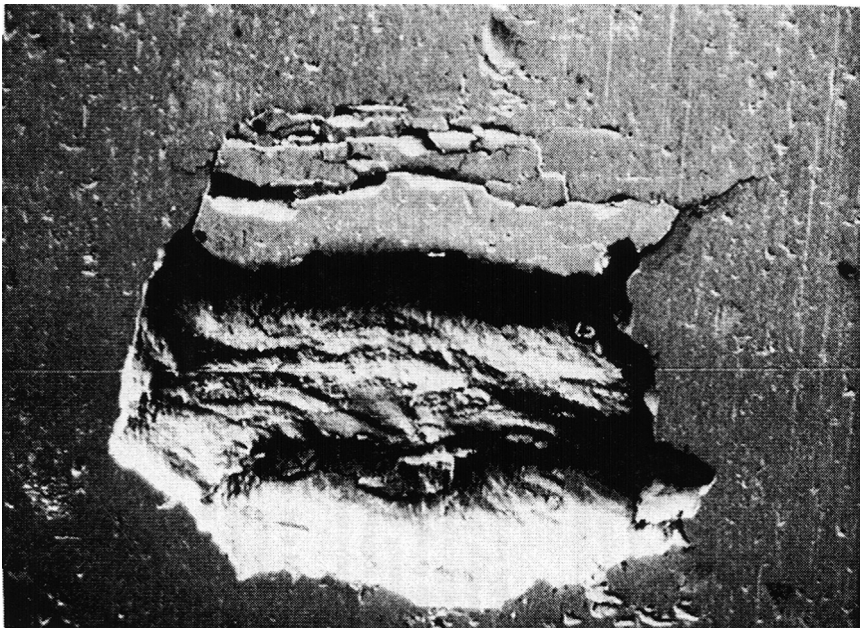

FIGURE 23. Subsurface nucleated spall on cylindrical bearing inner ring raceway. (Magnification × 50.)

brication. Visual gages are usually provided to facilitate checking for a continuous lubricant supply to all bearings in the system. In cases where separate bearings operate under different conditions of temperature, speed, and load, use of more than one system may be necessary to meet the correct lubrication needs of the individual bearings.

Circulating oil lubrication systems are most beneficial when bearings must be cooled continuously and when abrasive materials must be flushed away to assure safe operation. Circulating oil lubrication systems nearly always have filter and heat exchanger elements in addition to their oil reservoir and pump. They may also have a centrifuge or a sump for separating and removing foreign material, remote controls, warning devices, automatic cut-off switches, etc. These are particularly useful in meeting the special requirements of paper mills, lumber mills, steel mills, coal processing plants, and similar applications.

Oil mist lubrication systems use an air stream to provide oil to the bearings. The air pressure maintains a positive pressure within the bearing chamber which effectively prevents foreign matter from entering. The air flow can be regulated to produce minimum lubricant friction and the concomitant lubrication friction temperature effect. The air flow will not, however, provide significant cooling.

Air flowing out of a mist-lubricated bearing may discharge a fine oil vapor. This vapor may be objectionable, especially in the food and textile industries. In such cases, it is necessary to vent to other areas or provide air cleaning systems. Drainage of bearing reservoirs, provision for proper oil levels during bearing start-up, and timing of the mist flow must meet precise specifications. For this reason the system manufacturer should be relied upon to adjust the system for correct operation. Detailed information on lubricating systems is given in other chapters.

FAILURE ANALYSIS

Selection, application, and installation of rolling element bearings is based on subsurface nucleated fatigue. In the field, however, only 5 to 10% of the bearings removed from service are found to have developed this type of failure such as illustrated in Figure 23.

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Table 18
FAILURE MODES THAT LIMIT PERFORMANCE

Failure modes	Cause	Results
Brinelling	Dents in race or balls caused by impact or improper loading	Rough-running bearing surface origin fatigue
Cracked rings	Improper fits—abuse—cracked mounting	Catastrophic failure
Electric discharge	Flow of electric current through bearing	Rough-running bearing (fluting)
Fretting	Small amplitude motion of contact surfaces under load — causes indentation in race or localized material removal from mounting surfaces	Rough-running bearing — loss of bearing retention on shaft or in housing — stress concentration cracking
Galvanic action	Exposure of bearing to electrolyte (salt water in oil) and metal higher in galvanic series (copper, aluminum)	Pitting, leading to surface origin fatigue
Grease hardening/increased viscosity oil	Overheating of bearing — oil bleed loss — excessive bearing cavity ventilation — high ambient temperature	Excessive friction — wear — cage failure — surface origin fatigue
Lock-up	Temperature gradient due to shaft heating, causing loss of clearance	When steady state; preloading resulting in inner ring turning on shaft and severe shaft damage When intermittent: premature fatique failure of rollers and rings
Lubricant decomposition	Attack by lubricant decomposition products	Pitting, surface origin fatigue
Peeling-frosting	Low-viscosity lubricant, insufficient EHD film	Surface origin fatigue — increased clearance — rough operation
Race deformation	Overload and/or overheating	Destruction of geometry — increased clearance — cage failure — jamming
Rib wear and denting	Heavy impact loads — excessive thrust loads — abrasive contaminants — loss of lubricant or sparse lubricant	Loss of roller control where rib-guided — jamming of bearing — cage failure
Roller end wear	Skewing of cylindrical rollers under bearing misalignment — poor lubrication — abrasive contaminants	Slow deterioration leading to roller jamming — increased clearance
Rolling element banding	Excessive thrust load or overheating	Increase in internal clearance — rough-running after bearing stops and is started again
Rolling element scoring	Trapping of hard particles between retainer pocket and roller, causing deep cuts or scratches	Surface origin fatigue
Rusting	Exposure to humid or salt environment with insufficient protection by lubricant	Rough-running — surface origin fatigue
Smearing	Material transfer during skidding — loss of traction in load zone — very light loads — rapid acceleration	Rough-running bearing — surface origin fatigue
	Smearing of roller ends and guide flange	Overheating — loss of guidance — increase in axial float

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Table 18 (continued)
FAILURE MODES THAT LIMIT PERFORMANCE

Failure modes	Cause	Results
Wear of cages	Misalignment with nonaligning bearings — loss of lubricant or sparse lubricant — abrasive contaminants	Cage breakage — cage drop and jamming of bearing
Wear of rolling elements	Loss of or sparse lubricant — abrasive contamination	Jamming of bearing by debris — excessive internal clearance — retainer breakage

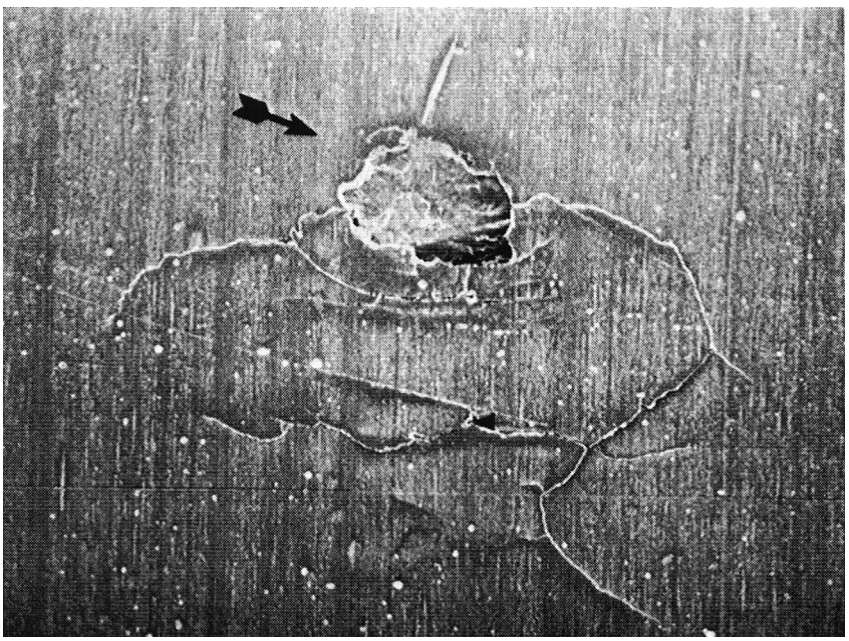

FIGURE 24. Scanning electron micrograph of surface nucleated spall.

Fatigue can often be induced by maldistribution of load in bearings due to varying stiffness of the mounting or support surfaces, housings, or shafts. Recognizing the sensitivity of rolling element bearing life to the variations in stress under the most heavily loaded rolling element (ball bearing life ~ $(1/\text{Stress})^{8\text{-}10}$, roller bearing life ~ $(1/\text{Stress})^{7\text{-}9}$), the designer must carefully consider the mounting, its stiffness, and the influence of mutual deflections of all components in the system. Distortions due to temperature distributions are equally important and transient conditions must be properly accounted for.

Damage commonly results from imposed loads which differ considerably from those anticipated in a machine design. Misalignment or fitting errors in mounting a bearing, misalignment or coupling faults between two machines, differential thermal expansion in a frame and shaft system, and rotor unbalance are among such factors. Simple visual or low power microscopic analysis of the ball paths in a ball bearings will frequently enable a useful evaluation of the magnitude and nature of these operating conditions.[30]

Table 18 lists failure modes that limit the performance of rolling element bearings. Several bearing companies have published similar lists and several volumes have been written on the subject. Of particular note is Reference 31. Detailed failure analysis should be correlated with the bearing company involved since their laboratory, background, and experience enable them to draw conclusions and make recommendations.

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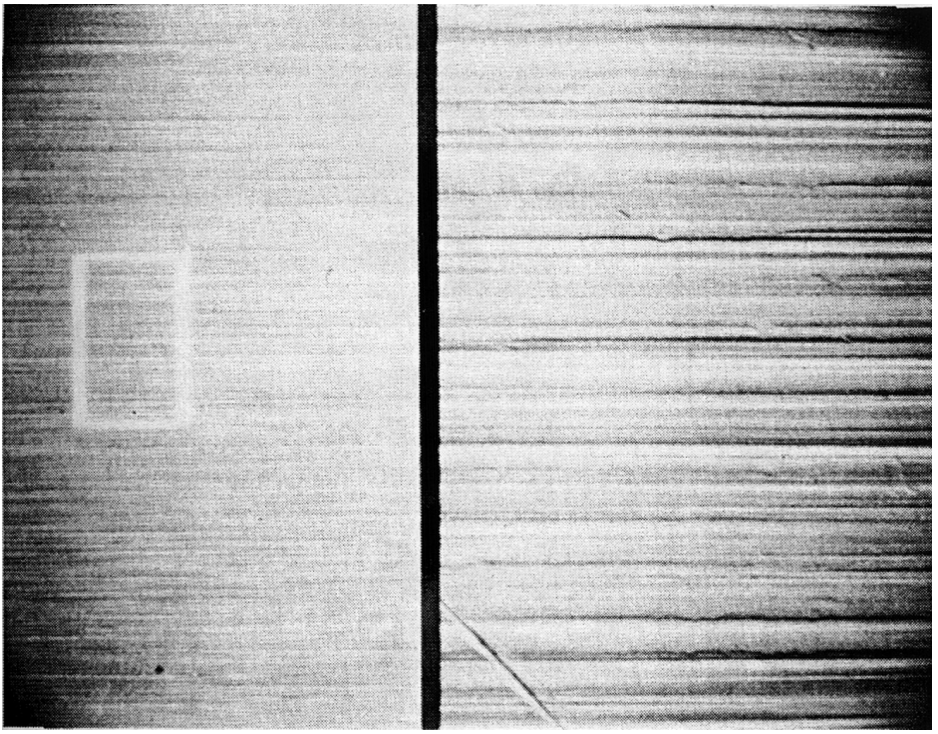

FIGURE 25. SEM of ground surface after running. (Magnifications ×□00 and 500.)

Bearings which have been grease-lubricated require special attention since they generally show surface effects which are the combination of many operating regimes. Grease lubrication in many bearings is a variable which depends on frequency of relubrication or on cyclic temperature variations to which the bearing is exposed. Many greased bearings operate with depleted films and significant wear will obliterate many original evidences of loading. Caution must be exercised against relying on the obvious conclusions while paying insufficient attention to the minor findings evidenced on careful examination.

Failure analysis has been greatly aided by utilization of the scanning electron microscope.[31,32] Small differences in the surface can indicate either the immediate condition of the bearing or some condition which has resulted from its previous operation. Figures 24 to 27 show scanning electron micrographs of bearing components which had been removed from service. Equally important is analysis of lubricants which can indicate the extent of contamination and deterioration. More recently, "ferrographic" analysis of filtrants or tailings from lubricants has become a valuable tool in monitoring transient bearing condition.

Since catastrophic failures have reduced value in aiding the troubleshooter, every effort must be made to look at units which have not failed completely, preferably a number of them with different periods of operation in order to detect and trace the incipient failure mode. Of particular importance is the observation of changes in surfaces, the lubricant, housing, and shaft as well as the bearing to correct outside influences that can cause early bearing failure.

In many instances, misalignment of sufficient magnitude to cause moment loading to run the rolling elements off the raceway induces a violent premature fatigue failure. Obviously, severe misalignment of tapered and cylindrical bearings must be corrected initially. Practical limits for misalignment are shown in Table 13.

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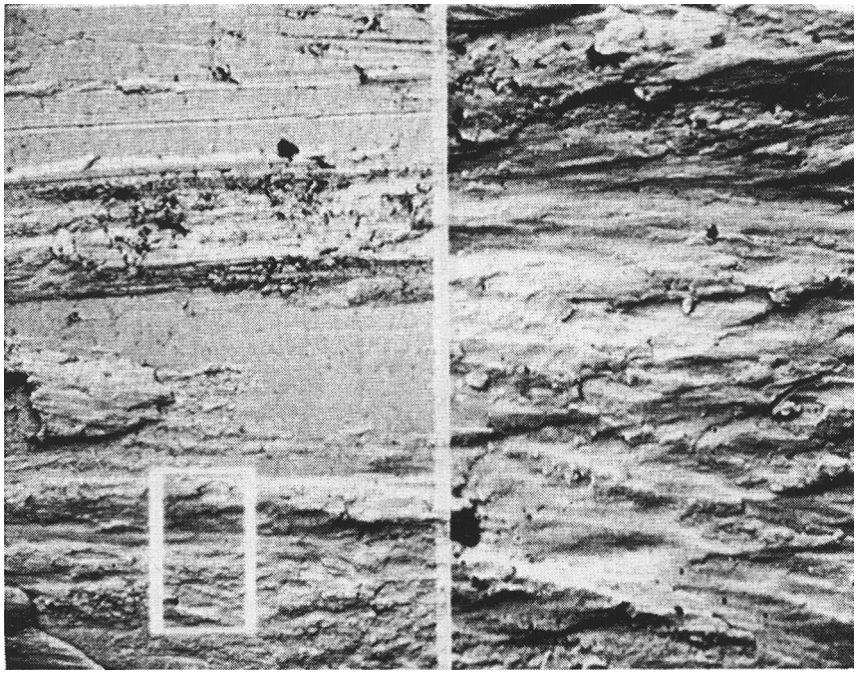

FIGURE 26. SEM of low-velocity galled roller. (Magnifications × 100 and 500.)

FIGURE 27. SEM of fretting wear in bore. (Magnification × 100.)

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